PATENT ABSTRACTS OF JAPAN

(11)Publication number : 2002-372319
(43)Date of publication of application : 26.12.2002

(51)Int.Cl. F25B 5/02
F25B 1/00
F25D 11/00
F25D 23/06

(21)Application number : 2001-184513
(22)Date of filing : 19.06.2001

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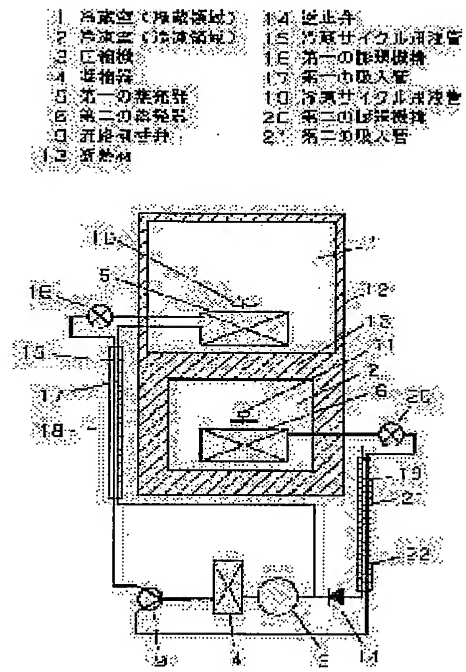
(54) REFRIGERATOR

(57)Abstract:
PROBLEM TO BE SOLVED: To provide heat absorption load constitution of heat insulation casing body to efficiently perform stable control of temperature during cooling operation of a refrigerator.
SOLUTION: A first evaporator 5 and a second evaporator 6 are respectively situated in a fresh food storage compartment 1 and a frozen food storage compartment freezer 2. The refrigerant circuit of the first evaporator 5 and the refrigerant circuit of the second vaporizer 6 are switched by a flow passage switching valve 9 for cooling. Since the heat absorption load amount of the frozen food storage compartment 2 during stable operation on the standard cooling condition of a refrigerator is equal to or less than the heat absorption load of the fresh food storage compartment 1, cooling operation time of the compartment 1 being high in refrigerating capacity can be maintained by restricting cooling operation time of the compartment 2 low in refrigerating capacity. Since an operation rate is prevented from being reduced to an extremely low value, control of the temperature of the compartment 1 is facilitated and a cooling loss during starting of a compressor 3 is suppressed and efficient operation is realized.



CLAIMS

[Claim(s)]
[Claim 1]A refrigerant circuit which is the refrigerator provided with a refrigeration field and a refrigerating area in an adiabatic box, has an evaporator in said refrigeration field and said refrigerating area, respectively, and pours a refrigerant to an evaporator of said refrigeration field at least, A refrigerator having provided a refrigerant circuit which pours a refrigerant to an evaporator of said refrigerating area, and making an endothermic burden of said refrigerating area at the time of stable operation on standard cooling conditions of a refrigerator into an endothermic burden of said refrigeration field below equivalent in what changes these refrigerant circuits and is cooled.
[Claim 2]The refrigerator according to claim 1, wherein an adiabatic wall of an adiabatic box was formed with a foamed heat insulating material and allocates a vacuum insulation material in said adiabatic wall of a refrigerating area.
[Claim 3]The refrigerator according to claim 1, wherein an adiabatic wall of an adiabatic box was formed with a foamed heat insulating material and allocates a vacuum insulation material in said adiabatic wall in 50 to 80% of range of outer packaging surface area.
[Claim 4]Are the refrigerator provided with a refrigeration field and a refrigerating area in an adiabatic box, have the first evaporator to said refrigeration field, have the second evaporator in said refrigerating area, and A compressor, While constituting a condenser, a flow path selector valve, a liquid tube for refrigeration cycles, said first evaporator, and the first suction pipe that carries out heat exchange to said liquid tube for refrigeration cycles from a closed loop, So that it may become in parallel with said liquid tube for refrigeration cycles, said first expansion mechanism, said first evaporator, and said first suction pipe A liquid tube for refrigerating cycles, The second expansion mechanism, said second evaporator, and the second suction pipe that carries out heat exchange to said liquid tube for refrigerating cycles, It is what performs mutually cooling of said refrigeration field and said refrigerating area independently by connecting a check valve and changing a flow of a refrigerant by said flow path selector valve, A refrigerator, wherein a power up makes resistance of said second expansion mechanism smaller than resistance at the time of stable operation on standard cooling conditions of a refrigerator.
[Claim 5]The refrigerator according to claim 4 in which a liquid tube for refrigeration cycles and a liquid tube for refrigerating cycles are characterized by an inside diameter being 0.8 mm or more.
[Claim 6]The refrigerator according to claim 4 in which a liquid tube for refrigeration cycles or a liquid tube for refrigerating cycles is formed two or more parallel liquid tubes, and said liquid tube is characterized by an inside diameter being 0.5 mm or more.
[Claim 7]The refrigerator according to any one of claims 4 to 6, wherein the first expansion mechanism and second expansion mechanism are the expansion valve installed in air in a warehouse, and an isolated portion.
[Claim 8]The first expansion mechanism or second expansion mechanism is formed by two or more capillaries which carry out heat exchange to the first suction pipe or the second suction pipe, The refrigerator according to claim 4 changing resistance by substituting said two or more capillaries for a liquid tube for refrigeration cycles, or a liquid tube for refrigerating cycles, and changing a channel of two or more capillaries.
[Claim 9]Have the following, and constitute said compressor, said condenser, said flow path selector valve, said first capillary, said third evaporator, and said third suction pipe from a closed loop, and. By connecting said second capillary so that it may become said first capillary and parallel, and changing a flow of a refrigerant to a capillary by said flow path selector valve, A refrigerator which uses said first capillary when opening said first air course and said second air course, and is characterized by changing a flow of a refrigerant using said second capillary when opening only the second air course.
It is the refrigerator provided with a refrigeration field and a refrigerating area, and is a compressor.
A condenser.
A flow path selector valve.
The first capillary, the second capillary, the third evaporator, the third suction pipe that carries out heat exchange to said first capillary and the second capillary, the first air course that carries out heat exchange of the air in said refrigeration field to said third evaporator, and the second air course that carries out heat exchange of the air in a refrigerating area to said third evaporator.

[Claim 10]A compressor is number-of-rotations good transformation, and, as for the first expansion mechanism and second expansion mechanism, the amount of diaphragms can change, Have an outside air temperature sensor which detects outdoor air temperature, and the amount of diaphragms of said first expansion mechanism and the second expansion mechanism is controlled so that a required refrigerant flow rate equivalent to a burden computed from outside air temperature which said outside air temperature sensor detected circulates; A refrigerator of nine given in any 1 paragraph from claim 4 controlling number of rotations of said compressor to become predetermined evaporating temperature from said required refrigerant flow rate.
[Claim 11]A refrigerator of ten given in any 1 paragraph from claim 4 which formed a receiver between a condenser and a flow path selector valve.

[Translation done.]

DETAILED DESCRIPTION

[Detailed Description of the Invention]
[0001]
[Field of the Invention]This invention relates to the refrigerator which attained efficient-ization by cooling cold storage and a freezer compartment independently with a separate evaporator.
[0002]
[Description of the Prior Art]In recent years, the schematic diagram of the refrigerator currently indicated by JP,11-148761,A as the conventional cooling cycle and an example of a refrigerator at drawing 13 considering cold storage and a freezer compartment as a thing about the refrigerator which has a separate evaporator is shown.
[0003]Cold storage and 2 are the first evaporator that allocate a freezer compartment and 3 in a compressor, and 4 was allocated by the condenser, and was allocated in the cold storage 1 5 1, and 6 is the second evaporator allocated in the freezer compartment 2.
[0004]7 is the first capillary allocated in the refrigerant circuit upstream of the first evaporator 5 that is an object for cold storage cooling, 8 is the second capillary allocated in the refrigerant circuit upstream of the second evaporator 6 that is an object for freezer compartment cooling, The first fan for making the cold storage 1 circulate through the flow path selector valve to which 9 changes the channel of a refrigerant, and the cold in which 10 carried out heat exchange to the first evaporator 5, The second fan for 11 to make the freezer compartment 2 circulating through the cold which carried out heat exchange to the second evaporator 6, and 12 are a refrigerator body and thermal insulation in which 13 controls the heat intruding to the interior of a room [open air].
[0005]About the refrigerator of the conventional example constituted as mentioned above, the operation is explained below.
[0006]Operation of a refrigerating cycle is performed as follows. The refrigerant first compressed by the compressor 3 is condensate-ized with the condenser 4. It is decompressed by the first capillary 7 or second capillary 8, and the condensed refrigerant flows into the first evaporator 5 and the second evaporator 6, and after evaporation evaporation is carried out, it is inhaled again to the compressor 3, respectively.
[0007]Each ** is cooled because the air of the cold storage 1 and the freezer compartment 2 carries out heat exchange to the first evaporator 5 and the second evaporator 6 which the refrigerant carried out evaporation evaporation and became low temperature comparatively with the first fan 10 and the second fan 11 and cold circulates.
[0008]Cooling down of a freezing refrigerator is performed as follows by the temperature detecting means and control means of each ** which are not illustrated.
[0009]Operation of a refrigerating cycle is performed until the compressor 3 will start and below a predetermined value will become, if each temperature detecting means of the cold storage 1 and the freezer compartment 2 detects the rise in heat beyond a predetermined value.
[0010]It flows only into the first evaporator 5, without a refrigerant flowing into the second evaporator 6 by the flow path selector valve 9, when the temperature detecting means of the cold storage 1 becomes beyond a predetermined value. Compared with the case where the temperature setting of the cold storage 1 is 0—15 ** to about 5 **, and the evaporating temperature at this time is operated with -25—30 ** evaporating temperature, operation of a compressor is performed by one 2 to 2.5 times the coefficient of performance of this.
[0011]When the temperature detecting means of the freezer compartment 2 becomes beyond a predetermined value, a refrigerant flows into the second evaporator 6 by the flow path selector valve 9, and cooling of the freezer compartment 2 is performed. As for the evaporating temperature at this time, the temperature setting of a freezer compartment is cooled at the usual evaporating temperature-25 ** to -30 ** to about -18 **.
[0012]The compressor 3 operates with a maximum engine speed to a power up, and is operating with the minimum engine speed at the time of the stable operation on the standard cooling conditions of a refrigerator.
[0013]Since the cold storage 1 and the freezer compartment 2 are repeated by turns as mentioned above and it cools, High evaporating temperature (0—20 **) is possible for the time of cold storage 1 cooling by circulating a refrigerant to the first evaporator independently, can make the compression ratio of the compressor 3 small, operate with a high coefficient of performance, and attain increase in efficiency, and. A temperature change is reduced by making small the difference of the room temperature of the cold storage 1, and evaporating temperature, and the temperature averaging of the cold storage 1 is aimed at. Further energy saving is performed by the compressor 3 quenching by operating with a maximum engine speed to a power up, operating with a minimum engine speed at the time of the stable operation on the standard cooling conditions of a refrigerator, and raising evaporating temperature.
[0014]Here, if evaporating temperature of -10 ** and the second evaporator 6 is made into -30 ** for the evaporating temperature of the first evaporator 5 and HFC134a is used as a refrigerant for example, the density of the refrigerant gas which evaporates with the first evaporator 5 will be about 2.3 times the density of the refrigerant gas which evaporates with the second evaporator 6. Even if it uses HC600a as a refrigerant similarly, it becomes about 2.2 times.
[0015]As a result, when usually making the same the number of rotations of the compressor 3 of cold storage cooling at the time of load, and freezer compartment cooling, the refrigerant amount which sets up resistance of the second capillary 8 twice [about] to the first capillary 7, and flows into the second evaporator 6 is made small, and evaporating temperature of -30 ** is realized. When usually changing the number of rotations of the compressor 3 of cold storage cooling at the time of load, and freezer compartment cooling, and it makes a refrigerant flow rate almost the same mostly by considering resistance of the first

capillary 7 and the second capillary 8 as the same and freezer compartment cooling is performed, it is also possible to raise number of rotations and to realize evaporating temperature of -30 **.
[0016]
[Problem(s) to be Solved by the Invention]However, when the rate of an endothermic load ratio of a refrigeration field is [especially the above-mentioned conventional composition] small in the refrigerator of high heat insulation performance with small endothermic load, while the operation time of a cold storage cooling cycle becomes extremely small and the temperature control of cold storage becomes difficult, The rate of the cooling loss at the time of compressor start became large, and there was a fault of operation efficient as a result becoming impossible.
[0017]This invention solves the conventional technical problem and it aims at realizing endothermic load composition of the adiabatic box which can perform efficiently temperature control stable at the time of cooling down of a refrigerator.
[0018]Since the number-of-rotations range of a compressor has a limit when the number of rotations of a compressor is lowered at the time of cold storage cooling cycle operation and it corresponds, The capability variable range at the time of freezer compartment cooling cycle operation is limited, and the problem from which the refrigerating capacity of the freezer compartment cooling cycle at the time of overload operation when load like [at a power up or the time of a defrosting return] as a result increases rapidly is not acquired enough arises.
[0019]At the time of overload operation when load like [at a power up or the time of a defrosting return] increases rapidly, immobilization of resistance of the capillary had the fault that it became difficult to make the refrigerating capacity of a freezer compartment cooling cycle with low refrigerant-gas density increase. In the refrigerator of the high heat insulation performance which aimed at energy saving, this is because synthetically high efficiency is obtained for the direction which optimizes the capability of a compressor, and resistance of a capillary according to remarkable low refrigerating capacity required at the time of the stable operation on the standard cooling conditions of a refrigerator.
[0020]For example, as shown in drawing 14, at the capillary A doubled with the refrigerant flow rate required for the endothermic burden of comparatively high outside air temperature, by comparatively low outside air temperature, the refrigerant flow rate more than needed will flow, the ratio of a refrigerant gas, i.e., the dryness fraction of a refrigerant, will increase as a result, and flow control will be performed automatically. At the capillary B doubled with the refrigerant flow rate required for the endothermic burden of comparatively low outside air temperature, although the adjustment cost by a refrigerant gas becomes small, by comparatively high outside air temperature, it becomes insufficient [a refrigerant amount].
[0021]Other purposes of this invention aim at providing a cooling function with it at the time of the overload operation at a power up, the time of a defrosting return, etc. [high efficiency and] [quick]
[0022]
[Means for Solving the Problem]A refrigerant circuit which the invention of this invention according to claim 1 is the refrigerator provided with a refrigeration field and a refrigerating area in an adiabatic box, has an evaporator in said refrigeration field and said refrigerating area, respectively, and pours a refrigerant to an evaporator of said refrigeration field at least, In what provides a refrigerant circuit which pours a refrigerant to an evaporator of said refrigerating area, changes these refrigerant circuits, and is cooled, Since it is a refrigerator making an endothermic burden of said refrigerating area at the time of stable operation on standard cooling conditions of a refrigerator into an endothermic burden of said refrigeration field below equivalent, Since it can prevent that refrigerating capacity can maintain cooling operation time of a large refrigeration field, and becomes the extreme low operating efficiency of 15% or less by controlling cooling operation time of a refrigerating area where refrigerating capacity is comparatively low, While temperature control of a refrigeration field becomes easy, a rate of a cooling loss at the time of compressor start is controlled, and operation efficient as a result can be attained.
[0023]Since the invention of this invention according to claim 2 is the refrigerator according to claim 1, wherein an adiabatic wall of an adiabatic box was formed with a foamed heat insulating material and allocates a vacuum insulation material in said adiabatic wall of a refrigerating area, While securing effective content volume, without thickening an adiabatic wall, cooling operation time of a refrigeration field where refrigerating capacity is large is maintainable.
[0024]Since the invention of this invention according to claim 3 is the refrigerator according to claim 1, wherein an adiabatic wall of an adiabatic box was formed with a foamed heat insulating material and allocates a vacuum insulation material in said adiabatic wall in 50 to 80% of range of outer packaging surface area, While securing effective content volume, without thickening an adiabatic wall, high cost performance is obtained by allocating a vacuum insulation material effectively.
[0025]The invention of this invention according to claim 4 is the refrigerator provided with a refrigeration field and a refrigerating area in an adiabatic box, and have the first evaporator to said refrigeration field, have the second evaporator in said refrigerating area, and A compressor, While constituting a condenser, a flow path selector valve, a liquid tube for refrigeration cycles, said first evaporator, and the first suction pipe that carries out heat exchange to said liquid tube for refrigeration cycles from a closed loop, So that it may become in parallel with said liquid tube for refrigeration cycles, said first expansion mechanism, said first evaporator, and said first suction pipe A liquid tube for refrigerating cycles, The second expansion mechanism, said second evaporator, and the second suction pipe that carries out heat exchange to said liquid tube for refrigerating cycles, It is what performs mutually cooling of said refrigeration field and said refrigerating area independently by connecting a check valve and changing a flow of a refrigerant by said flow path selector valve, Since it is a refrigerator, wherein a power up makes resistance of said second expansion mechanism smaller than resistance at the time of stable operation on standard cooling conditions of a refrigerator, Obtaining high evaporating temperature at the time of the same refrigeration field cooling as the former at the time of stable operation on standard cooling conditions of a refrigerator, and maintaining an energy-saving cycle by reducing circulation of a gas refrigerant at the time of refrigerating area cooling, and obtaining a low refrigerant flow rate corresponding to low loading. At the time of overload operation, such as a power up, at the time of refrigerating area cooling, it is considered as the amount of high refrigerant circulation equivalent to the time of refrigeration field cooling, and quenches efficiently by considering it as evaporating temperature used as heat exchanging capacity corresponding to the amount of refrigerant circulation.
[0026]Since a liquid tube for refrigeration cycles and a liquid tube for refrigerating cycles are the refrigerators according to claim 4, wherein an inside diameter is 0.8 mm or more, the invention of this invention according to claim 5, Make it into the amount of high refrigerant circulation equivalent to the time of refrigeration field cooling at the time of refrigerating area cooling at the time of overload operation, such as a power up, maintaining energy saving at the time of stable operation on standard cooling conditions of a refrigerator, and. It quenches efficiently by considering it as evaporating temperature used as heat exchanging capacity corresponding to the amount of refrigerant circulation. Volume of a refrigerant which stagnates in a liquid tube for refrigeration cycles or a liquid tube for refrigerating cycles is controlled in a small quantity, it is stabilized and control of flow of

an expansion mechanism can be performed.
[0027]The invention of this invention according to claim 6 is formed with two or more liquid tubes with which a liquid tube for refrigeration cycles or a liquid tube for refrigerating cycles was parallel, and since said liquid tube is the refrigerator according to claim 4, wherein an inside diameter is 0.5 mm or more, Maintaining energy saving due to a raise in evaporating temperature at the time of refrigeration field cooling, and the fall of entrance refrigerant dryness of an expansion mechanism for refrigerating area cooling at the time of stable operation on standard cooling conditions of a refrigerator. In addition to the ability to do quenching for a power up efficiently, the heat exchange length of a suction pipe and a liquid tube is shortened, and volume of a refrigerant which stagnates in a liquid tube for refrigeration cycles or a liquid tube for refrigerating cycles is controlled in a small quantity, it is stabilized and control of flow of an expansion mechanism can be performed.
[0028]Since the invention of this invention according to claim 7 is a refrigerator of six given in any 1 paragraph from claim 4, wherein the first expansion mechanism and second expansion mechanism are the expansion valve installed in air in a warehouse, and an isolated portion, Maintaining energy saving due to a raise in evaporating temperature at the time of refrigeration field cooling, and the fall of entrance refrigerant dryness of an expansion mechanism for refrigerating area cooling at the time of stable operation on standard cooling conditions of a refrigerator. quenching can be efficiently done for a power up — in addition, it can control that a refrigerant is revealed to the interior of a room at the time of refrigerant disclosure.
[0029]The invention of this invention according to claim 8 forms the first expansion mechanism or second expansion mechanism by two or more capillaries which carry out heat exchange to the first suction pipe or the second suction pipe, Since it is the refrigerator according to claim 4 changing resistance by substituting said two or more capillaries for a liquid tube for refrigeration cycles, or a liquid tube for refrigerating cycles, and changing a channel of two or more capillaries, Maintaining energy saving due to a raise in evaporating temperature at the time of refrigeration field cooling, and the fall of entrance refrigerant dryness of an expansion mechanism for refrigerating area cooling at the time of stable operation on standard cooling conditions of a refrigerator. In addition to the ability to do quenching for a power up efficiently, refrigerant filling quantity can be reduced by substituting for a liquid tube a capillary which is small volume.
[0030]The invention of this invention according to claim 9 is a refrigeration field and a refrigerating area the refrigerator which it had, and A compressor, A condenser, a flow path selector valve, the first capillary, and the second capillary, The third evaporator and the third suction pipe that carries out heat exchange to said first capillary and the second capillary, The first air course that carries out heat exchange of the air in said refrigeration field to said third evaporator, Have the second air course that carries out heat exchange of the air in a refrigerating area to said third evaporator, and constitute said compressor, said condenser, said flow path selector valve, said first capillary, said third evaporator, and said third suction pipe from a closed loop, and. By connecting said second capillary so that it may become said first capillary and parallel, and changing a flow of a refrigerant to a capillary by said flow path selector valve, Since it is a refrigerator changing a flow of a refrigerant using said second capillary when using said first capillary when opening said first air course and said second air course, and opening only the second air course, Can quench efficiently by performing by turns cooling down which cools a refrigeration field and a refrigerating area simultaneously by the first small capillary of resistance at the time of overloads, such as a power up, and cooling down which cools only a refrigerating area, and. At the time of stable operation on standard cooling conditions of a refrigerator, refrigeration field cool time is lengthened by cooling a refrigeration field and a refrigerating area simultaneously, and a temperature change in a refrigeration field can be controlled.
[0031]A compressor of the invention of this invention according to claim 10 is number-of-rotations good transformation, The amount of diaphragms can change and the first expansion mechanism and second expansion mechanism have an outside air temperature sensor which detects outdoor air temperature, The amount of diaphragms of said first expansion mechanism and the second expansion mechanism is controlled so that a required refrigerant flow rate equivalent to a burden computed from outside air temperature which said outside air temperature sensor detected circulates, From claim 4 controlling number of rotations of said compressor to become predetermined evaporating temperature from said required refrigerant flow rate, since it is a refrigerator of nine given in any 1 paragraph, At the time of overloads, such as a power up, it can quench efficiently, and it becomes the evaporating temperature which can always acquire heat exchanging capacity corresponding to a refrigerant flow rate, and cools efficiently using the maximum capacity of a refrigerating cycle.
[0032]Since the invention of this invention according to claim 11 is a refrigerator of ten given in any 1 paragraph from claim 4 which formed a receiver between a condenser and a flow path selector valve, A temporary shortage of the amount of refrigerant circulation when changing from refrigerating area cooling of the amount of low refrigerant circulation to refrigeration field cooling of the amount of high refrigerant circulation is relieved, and it can shift to an efficient cycle of cold storage cooling at an early stage.
[0033]
[Embodiment of the Invention]Embodiment 1 by this invention is described referring to drawings. About a conventional example and an identical configuration, identical codes are attached and detailed explanation is omitted.
[0034](Embodiment 1) Drawing 1 is a schematic diagram of the cooling cycle by the embodiment of the invention 1, and a refrigerator.
[0035]The endothermic burden of the refrigeration field where the endothermic burden of the refrigerating area which consists of the freezer compartment 2 at the time of the stable operation on the standard cooling conditions of a refrigerator in drawing 1 consists of the cold storage 1 is the same in abbreviation.
[0036]The liquid tube for refrigeration cycles in which a check valve circulates 14 and a refrigerant circulates 15 in drawing 1 at the time of cold storage cooling, The first suction pipe to which 16 connects the first expansion mechanism to and 17 connects the first evaporator 5 and compressor 3, The first heat exchanging part to which the liquid tube 15 for refrigeration cycles and the first suction pipe 17 carry out heat exchange of 18, The liquid tube for refrigerating cycles in which a refrigerant circulates 19 at the time of freezer compartment cooling, the second expansion mechanism that is flow good transformation 20, the second suction pipe that connects the second evaporator 6 and compressor 3 21, and 22 are the second heat exchanging part in which the liquid tube 19 for refrigerating cycles and the second suction pipe 21 carry out heat exchange.
[0037]About the refrigerator constituted as mentioned above, the operation is explained below.
[0038]If temperature inside is detected by the temperature-inside sensor of the cold storage 1 which is not illustrated at the time of cooling of the cold storage 1 and it becomes more than prescribed temperature, a refrigerant is compressed by operation of the compressor 3, and the refrigerant of the compressed high temperature high pressure will be condensed by being cooled with the condenser 4, and will flow into the flow path selector valve 9. The refrigerant circulates to the liquid tube 15 for refrigeration cycles from the flow path selector valve 9 controlled to circulate an outlet side to the liquid tube 15 for refrigeration

cycles. When it passes along the liquid tube 15 for refrigeration cycles, by the first heat exchanging part 18, heat exchange will be carried out to the first suction pipe 17, it will be cooled, and a refrigerant will be in a supercooling state, and is sent to the first expansion mechanism 16. and a refrigerant serves as low temperature of evaporating temperature higher than the evaporating temperature at the time of cooling of the freezer compartment 2 by it being alike according to the first expansion mechanism 16, being decompressed, and evaporating, and flows through the first evaporator 5. At this time, it is cooled by carrying out heat exchange to the first evaporator 5 that became low temperature by the operation of the first fan 10, and the air in the cold storage 1 circulates, and performs cooling in the cold storage 1. And the refrigerant in the first evaporator 5 circulates a dryness fraction with increase, it becomes saturated gas and the entrance of the first suction pipe 17 is reached at the exit of the first evaporator 5. When it passes along the first suction pipe 17, this refrigerant is heated by carrying out heat exchange to the hot liquid tube 15 for refrigeration cycles by the first heat exchanging part 18, serves as moderate gas, and is inhaled by the compressor 3. Although the second evaporator 6 for cooling of the freezer compartment 2 is a room temperature grade of the freezer compartment 2 and is lower than the evaporating pressure of the first evaporator 5 at this time, the back run of the refrigerant is prevented by the check valve 14.

[0039]If temperature inside is detected by the temperature-inside sensor of the freezer compartment 2 which is not illustrated at the time of cooling of the freezer compartment 2 and it becomes more than prescribed temperature, a refrigerant is compressed by operation of the compressor 3, and the refrigerant of the compressed high temperature high pressure will be condensed by being cooled with the condenser 4, and will flow into the flow path selector valve 9. The refrigerant circulates to the liquid tube 19 for refrigerating cycles from the flow path selector valve 9 controlled to circulate an outlet side to the liquid tube 19 for refrigerating cycles. When it passes along the liquid tube 19 for refrigerating cycles, by the second heat exchanging part 22, heat exchange will be carried out to the second suction pipe 21, it will be cooled, and a refrigerant will be in a supercooling state, and is sent to the second expansion mechanism 20. and a refrigerant serves as low evaporating temperature by it being alike according to the second expansion mechanism 20, being decompressed, and evaporating, and flows through the second evaporator 6. At this time, it is cooled by carrying out heat exchange to the second evaporator 6 that became low temperature by the operation of the second fan 11, and the air in the freezer compartment 2 circulates, and performs cooling in the cold storage 1. And the refrigerant in the first evaporator 5 circulates a dryness fraction with increase, it becomes saturated gas and the entrance of the second suction pipe 21 is reached at the exit of the second evaporator 6. When it passes along the second suction pipe 21, this refrigerant is heated by carrying out heat exchange to the hot liquid tube 19 for refrigerating cycles by the second heat exchanging part 22, serves as moderate gas, and is inhaled by the compressor 3.

[0040]In the time of stable operation [in / here / the standard cooling conditions of a refrigerator], The compressor 3 is operated with a minimum engine speed, and the second expansion mechanism 20 is adjusted so that resistance may be twice to the first expansion mechanism 16, and it is controlling the evaporating temperature at the time of cooling of -30 ** and the cold storage 1 for the evaporating temperature at the time of freezer compartment 2 cooling at -15 **. Since the amount of refrigerant circulation at the time of cold storage 1 cooling will be the twice [about] at the time of freezer compartment 2 cooling at this time, the cooling operation time of the cold storage 1 can be adjusted to the refrigerating capacity corresponding to the endothermic burden ratio of a refrigeration field and a refrigerating area by considering it as about 1 of the cooling operation time of the freezer compartment 2 / twice.

[0041]If the compressor 3 which has the refrigerating capacity in a suitable minimum engine speed is selected at this time, the compressor 3 can be run continuously mostly, changing cooling down of the cold storage 1, and cooling down of the freezer compartment 2. In this case, as for the operating efficiency of the cold storage 1, the operating efficiency of the freezer compartment 2 will be about 67% about 33% to 100% of the total operating efficiency. Since the loss which changes a refrigerant passage will become large if cooling down of the cold storage 1 and the freezer compartment 2 is changed frequently, it is desirable to make one cycle of the cooling + cooling shut down of the cooling + freezer compartment 2 of the cold storage 1 into 50 to 100 minutes. At this time, as for the operation time in 1 cycle of the cold storage 1, the operation time in 1 cycle of the freezer compartment 2 will be about 33 - 67 minutes for about 17 - 33 minutes. As a result, both cooling down of the cold storage 1 and the freezer compartment 2 can perform satisfactory efficient operation.

[0042]If the operating efficiency of the cold storage 1 or the freezer compartment 2 will be 15% or less here, while control of a temperature change will become difficult, If the operation time of the cold storage 1 or the freezer compartment 2 becomes 10 or less minutes, immediately after a refrigerant passage change or compressor start, the rate of the cooling loss which operates a compressor after the refrigerants in the first evaporator 5 have run short will become large, and efficient operation will become difficult. In the case of the system which changes a refrigerating area and a refrigeration field and is cooled, there is a tendency for the operation time of cooling down of the refrigeration field where the refrigerating capacity per unit time is high to become short, and the design of the endothermic burden of a refrigerating area and a refrigeration field becomes important.

[0043]It is desirable to make ideally the endothermic burden of the refrigerating area which consists of the freezer compartment 2 at the time of the stable operation on the standard cooling conditions of a refrigerator into about 1 of the endothermic burden of the refrigeration field which consists of the cold storage 1 / twice. In this case, since both the operating efficiency of the cold storage 1 and the freezer compartment 2 can also adjust operation time in 25 to 50 minutes about 50%, the problem of an operating efficiency fall or an operation-time fall is not produced. If the endothermic burden of the refrigerating area which consists of the freezer compartment 2 at the time of the stable operation on the standard cooling conditions of a refrigerator on the other hand will be about 3 times of the endothermic burden of the refrigeration field which consists of the cold storage 1, About 14%, operation time will also be 7 to 14 minutes, and it becomes difficult control of a temperature change and to control the operating efficiency of the cold storage 1 of a refrigerant passage change loss. Therefore, as for the endothermic burden of the refrigerating area at the time of the stable operation on the standard cooling conditions of a refrigerator, about 2 times is desirable from [of the endothermic burden of a refrigeration field] 1/2.

[0044]As mentioned above, by making the same in the endothermic burden of the refrigeration field which consists of the cold storage 1, and abbreviation the endothermic burden of the refrigerating area which consists of the freezer compartment 2 at the time of the stable operation on the standard cooling conditions of a refrigerator, The operation time of the cold storage 1 can be secured and the temperature change of the cold storage 1 and the rate of the cooling loss at the time of a change can be controlled.

[0045](Embodiment 2) Embodiment 2 by this invention is described, referring to drawings. About Embodiment 1, an identical configuration, and an operation, identical codes are attached and detailed explanation is omitted.

[0046]Drawing 2 is a schematic diagram of the cooling cycle by the embodiment of the invention 2, and a refrigerator.

[0047]In drawing 2, the thermal insulation 13 is the urethane insulation of thermal conductivity 0.015 W/mK by which normal use

is carried out, and 40 is a vacuum insulation material of the high heat insulation performance whose thermal conductivity is 0.003 W/mK, and it has covered about 50% of outer packaging surface area with the vacuum insulation material 40. And the endothermic burden of the refrigeration field where the endothermic burden of the refrigerating area which consists of the freezer compartment 2 at the time of the stable operation on the standard cooling conditions of a refrigerator consists of 13W and the cold storage 1 is 27W. The vacuum insulation material 40 carries out decompression deaeration, and becomes the inside which is indicated by the JP,60-146994,A gazette from the thermal insulation pack which wrapped the outside in the bag which is not in breathability, for example.
[0048]About the refrigerator constituted as mentioned above, the operation is explained below.
[0049]In the cold storage 1, at the time of the stable operation on the standard cooling conditions of a refrigerator, heat invades through the thermal insulation 13 from the open air. And the quantity of heat which invades from this open air is removed by operating the compressor 3 and evaporating a refrigerant with the first evaporator 5, and keeps the inside of the cold storage 1 at 5 **. In the freezer compartment 2, heat invades indoors through the thermal insulation 13 and the vacuum insulation material 40 from the open air. And the quantity of heat which invades from this open air is removed by operating the compressor 3 and evaporating a refrigerant with the second evaporator 6, and keeps the inside of the freezer compartment 2 at -20 **. At this time, the endothermic burden of the refrigerating area which consists of the freezer compartment 2 according to the adiabatic efficiency of the vacuum insulation material 40 can be controlled about 1 of the endothermic burden of the refrigeration field which consists of the cold storage 1 / twice, and the operating efficiency of the cold storage 1 and the freezer compartment 2 can be designed to about 50%.
[0050]The wall thickness at the time of laminating the case where only the usual thermal insulation 13 is used for heat insulation of the periphery of the freezer compartment 2 here, the thermal insulation 13, and the vacuum insulation material 40 is shown in (Table 1).
[0051]
[Table 1]

	合計厚さ	ウレタン断熱材		真空断熱材		熱通過率
		厚さ	熱伝導率	厚さ	熱伝導率	
	mm	mm	W/mK	mm	W/mK	W/m2K
実施の形態2	64	32	0.015	32	0.003	0.078
従来例	193	193	0.015	0	0.003	0.078

[0052]In (Table 1), a burden is an endothermic burden which invades in the freezer compartment 2 through the thermal insulation 13 or the vacuum insulation material 40 from the 25 ** open air or the cold storage 1. As shown in (Table 1), the freezer compartment 2 can make invasion quantity of heat extremely small also in the state where wall thickness is thin, by using the vacuum insulation material 40, and can be designed by the same wall thickness as the cold storage 1 with much invasion quantity of heat.
[0053]As mentioned above, by using a vacuum insulation material for heat insulation of the periphery of the freezer compartment 2, By making the endothermic burden of the refrigerating area which consists of the freezer compartment 2 at the time of the stable operation on the standard cooling conditions of a refrigerator into 1/2 twice the endothermic burden of the refrigeration field which consists of the cold storage 1, maintaining thin wall thickness, The operation time of the cold storage 1 can be secured and the temperature change of the cold storage 1 and the rate of the cooling loss at the time of a change can be controlled.
[0054]Although the vacuum insulation material 40 covered about 50% of the outer packaging surface area of the refrigerator in this embodiment, 50 to 80% of coverage is desirable. When coverage is smaller than 50%, while being unable to cover the whole refrigerating area but reduction of endothermic load being difficult, when coverage is larger than 80%, in the usual thermal insulation 13, in an outer packaging corner etc., the fall of light-gage next door strength of structure poses a problem by the matching of the vacuum insulation material 40. It is more desirable for the vacuum insulation material 40 to make plane shape thermal insulation 13 of the upper part of the machinery room in which the compressor 3 grade is installed, and the boundary part of the freezer compartment 2 in order to tend to install the direction of a flat-surface part.
[0055](Embodiment 3) Embodiment 3 by this invention is described, referring to drawings. About the former, an identical configuration, and operation, identical codes are attached and detailed explanation is omitted.
[0056]Drawing 3 is a schematic diagram of the cooling cycle by the embodiment of the invention 3, and a refrigerator.
[0057]The liquid tube for refrigeration cycles in which a check valve circulates 14 and a refrigerant circulates 15 in drawing 3 at the time of cold storage cooling, The first suction pipe to which 16 connects the first expansion mechanism to and 17 connects the first evaporator 5 and compressor 3, The first heat exchanging part to which the liquid tube 15 for refrigeration cycles and the first suction pipe 17 carry out heat exchange of 18, The liquid tube for refrigerating cycles in which a refrigerant circulates 19 at the time of freezer compartment cooling, the second expansion mechanism that is flow good transformation 20, the second suction pipe that connects the second evaporator 6 and compressor 3 21, and 22 are the second heat exchanging part in which the liquid tube 19 for refrigerating cycles and the second suction pipe 21 carry out heat exchange.
[0058]About the refrigerator constituted as mentioned above, the operation is explained below.
[0059]If temperature inside is detected by the temperature-inside sensor of the cold storage 1 which is not illustrated at the time of cooling of the cold storage 1 and it becomes more than prescribed temperature, a refrigerant is compressed by operation of the compressor 3, and the refrigerant of the compressed high temperature high pressure will be condensed by being cooled with the condenser 4, and will flow into the flow path selector valve 9. The refrigerant circulates to the liquid tube 15 for refrigeration cycles from the flow path selector valve 9 controlled to circulate an outlet side to the liquid tube 15 for refrigeration cycles, When it passes along the liquid tube 15 for refrigeration cycles, by the first heat exchanging part 18, heat exchange will be carried out to the first suction pipe 17, it will be cooled, and a refrigerant will be in a supercooling state, and is sent to the first expansion mechanism 16. and a refrigerant serves as low temperature of evaporating temperature higher than the evaporating temperature at the time of cooling of the freezer compartment 2 by it being alike according to the first expansion mechanism 16, being decompressed, and evaporating, and flows through the first evaporator 5. At this time, it is cooled by carrying out heat exchange to the first evaporator 5 that became low temperature by the operation of the first fan 10, and the air in the cold storage 1 circulates, and performs cooling in the cold storage 1. And the refrigerant in the first evaporator 5 circulates a dryness fraction with increase, it becomes saturated gas and the entrance of the first suction pipe 17 is reached at the exit of the first

evaporator 5. When it passes along the first suction pipe 17, this refrigerant is heated by carrying out heat exchange to the hot liquid tube 15 for refrigeration cycles by the first heat exchanging part 18, serves as moderate gas, and is inhaled by the compressor 3. Although the second evaporator 6 for cooling of the freezer compartment 2 is a room temperature grade of the freezer compartment 2 and is lower than the evaporating pressure of the first evaporator 5 at this time, the back run of the refrigerant is prevented by the check valve 14.

[0060]If temperature inside is detected by the temperature-inside sensor of the freezer compartment 2 which is not illustrated at the time of cooling of the freezer compartment 2 and it becomes more than prescribed temperature, a refrigerant is compressed by operation of the compressor 3, and the refrigerant of the compressed high temperature high pressure will be condensed by being cooled with the condenser 4, and will flow into the flow path selector valve 9. The refrigerant circulates to the liquid tube 19 for refrigerating cycles from the flow path selector valve 9 controlled to circulate an outlet side to the liquid tube 19 for refrigerating cycles, When it passes along the liquid tube 19 for refrigerating cycles, by the second heat exchanging part 22, heat exchange will be carried out to the second suction pipe 21, it will be cooled, and a refrigerant will be in a supercooling state, and is sent to the second expansion mechanism 20. and a refrigerant serves as low evaporating temperature by it being alike according to the second expansion mechanism 20, being decompressed, and evaporating, and flows through the second evaporator 6. At this time, it is cooled by carrying out heat exchange to the second evaporator 6 that became low temperature by the operation of the second fan 11, and the air in the freezer compartment 2 circulates, and performs cooling in the cold storage 1. And the refrigerant in the first evaporator 5 circulates a dryness fraction with increase, it becomes saturated gas and the entrance of the second suction pipe 21 is reached at the exit of the second evaporator 6. When it passes along the second suction pipe 21, this refrigerant is heated by carrying out heat exchange to the hot liquid tube 19 for refrigerating cycles by the second heat exchanging part 22, serves as moderate gas, and is inhaled by the compressor 3.

[0061]While usually operating the compressor 3 with a minimum engine speed here at the time of operation, the second expansion mechanism 20 is adjusted so that resistance may be twice [about] to the first expansion mechanism 16, and is controlling the evaporating temperature at the time of cooling of -30 ** and the cold storage 1 for the evaporating temperature at the time of cooling of the freezer compartment 2 at -15 **.

[0062]And in a power up, the compressor 3 is operated with a maximum engine speed, and resistance of the second expansion mechanism 20 is controlled to become resistance and the equivalent grade of the first expansion mechanism 16. as a result, the increase in until comparable can be carried out with the refrigerant flow rate at the time of cold storage 1 cooling, and the refrigerant flow rate at the time of freezer compartment 2 cooling can be quenched. It is more desirable to set it as -20--30 ** at this time, since the evaporating temperature of the first evaporator 5 and the second evaporator 6 corresponds to the refrigerant flow rate which increased. Evaporating temperature rises at the same time a refrigerant flow rate will increase, if it resists small more than this, and since the heat exchange temperature gradient in an evaporator becomes small, the refrigerant in an evaporator cannot be evaporated and it becomes useless. Since freezer compartment 2 cooling and cold storage 1 cooling use the maximum capacity of a cooling system, they can make the shortest pulldown time from after powering on.

[0063]At the time of overloads, such as operation after defrosting the second evaporator 6, even if it controls resistance of the second expansion mechanism 20 like a power up, the effect which quenches the freezer compartment 2 is acquired.

[0064]As mentioned above, at the time of overloads, such as a power up, quenching can be efficiently done by controlling resistance of the second expansion mechanism 20 to become resistance and the equivalent grade of the first expansion mechanism 16.

[0065](Embodiment 4) Embodiment 4 by this invention is described, referring to drawings. About Embodiment 3, an identical configuration, and operation, identical codes are attached and detailed explanation is omitted.

[0066]The feature of the composition in this embodiment is that it formed 1.2 mm in inside diameter, 2.4 m in length, and the liquid tube 19 for refrigerating cycles for the liquid tube 15 for refrigeration cycles with the copper pipe with a smooth inner surface 0.8 mm in inside diameter, and 2.4 m in length. The liquid tube 15 for refrigeration cycles, and the liquid tube 19 for refrigerating cycles, While supplying the refrigerant liquefied with the condenser 4 to the first expansion mechanism 16 and second expansion mechanism 20, respectively, in the first heat exchanging part 18 and second heat exchanging part 22, heat exchange is carried out to the first suction pipe 17 and the second suction pipe 21, respectively.

[0067]Generally, although a copper narrow diameter pipe 3-4 mm in inside diameter is used, the liquid tube 15 for refrigeration cycles, and the liquid tube 19 for refrigerating cycles, Since the amount of liquid cooling intermediation held inside becomes large with 10-20g when using the inflammable refrigerant of R600a or R290 grade, narrow diameter-ization is desired from a viewpoint of reducing an operating refrigerant amount. According to this embodiment, while clarifying the limit of narrow-diameter-izing, the optimal amount of inside diameters in consideration of the time of usual operation of the refrigerator carrying a change system and a power up is proposed.

[0068]Drawing 4 is a P-h diagram of the cooling cycle by this embodiment. It is the cold storage cooling cycle of the power up of high outside air temperature with the largest circulating load which was shown by drawing 4.

[0069]The state of a refrigerant [in / on drawing 4 and / in A / the entrance of the liquid tube 15 for refrigeration cycles], The state of a refrigerant [in / in B / the entrance of the first expansion mechanism 16], the state of a refrigerant [in / C is an exit of the first expansion mechanism 16, and / the entrance of the first evaporator 5], D is an exit of the first evaporator 5, the state of the refrigerant in the entrance of the first suction pipe 17 and E are the exits of the first suction pipe 17, it is in the state of the refrigerant in the inhalation section of the compressor 3, and heat exchange of the liquid tube 15 for refrigeration cycles and the first suction pipe 17 is carried out 100% by the first heat exchanging part 18. Thereby, the difference of the enthalpy of the inhalation section of the compressor 3 and the enthalpy of the entrance of the first suction pipe 17 becomes equal to the difference of the enthalpy of the entrance of the liquid tube 15 for refrigeration cycles, and the enthalpy of the entrance of the first expansion mechanism 16. That is, the enthalpy difference of E and D is equal to the enthalpy difference of A and B.

[0070]Since a narrow diameter pipe 1.2 mm in inside diameter and 2.4 m in length is being used for the liquid tube 15 for refrigeration cycles of this embodiment, it can hold down the volume of the refrigerant R600a which circulates the inside of a pipe to 2-3g and the minimum quantity. However, the pressure loss by pipe internal resistance arises for a narrow diameter pipe, and as the B point showed, the pressure of the refrigerant in the entrance of the first expansion mechanism 16 declines from the pressure of the entrance of the liquid tube 15 for refrigeration cycles shown by the A point. Although a B point is in a supercooling region and fault is not produced in operation of the expansion mechanism 16 in this inside diameter, if an inside diameter is extracted to 0.8 mm, the pressure loss by pipe internal resistance will arise, and as B1 point of drawing 4 showed on condition of ************ of high outside air temperature which shows the largest circulating load, 0 ** of supercooling will be

in a last-minute state. If an inside diameter is furthermore made small, while it will shift to a two-phase region as a pressure loss increases and B-2 point of drawing 4 showed and operation of the expansion mechanism 16 will become unstable, the problem to which the resistance of the expansion mechanism 16 on appearance increases, and a refrigerant flow rate falls remarkably occurs.
[0071]Since the liquid tube 19 for refrigerating cycles of this embodiment is using a narrow diameter pipe 0.8 mm in inside diameter, and 2.4 m in length similarly, while being able to hold down the volume of the refrigerant R600a which circulates the inside of a pipe to 1g or less and the minimum quantity, A refrigerant can be circulated without shifting to a two-phase region also on the conditions of the power up of high outside air temperature which show the largest circulating load. At the time of usual operation which enlarges resistance of the second expansion mechanism 20 and controls it, the exit of the liquid tube 19 for refrigerating cycles will be in the state of a supercooling region like a refrigeration cycle.
[0072]As mentioned above, while being able to do quenching efficiently by controlling resistance of the second expansion mechanism to become resistance and the equivalent grade of the first expansion mechanism at the time of overloads, such as a power up, Holding down very much the volume of the refrigerant which circulates the inside of a pipe to a small quantity, it is stabilized and control of flow of the first expansion mechanism and the second expansion mechanism can be performed because the inside diameter of the liquid tube for refrigeration cycles and the liquid tube for refrigerating cycles shall be 0.8 mm or more.
[0073](Embodiment 5) Embodiment 5 by this invention is described, referring to drawings. About Embodiment 4, an identical configuration, and operation, identical codes are attached and detailed explanation is omitted.
[0074]The cooling cycle according [drawing 5] to the embodiment of the invention 4 and the schematic diagram of a refrigerator, and drawing 6 are the strabism sectional views of an important section. In drawing 5 and drawing 6, 23 and 24 are the first liquid tube and second liquid tube.
[0075]The feature of the composition in this embodiment is that the channel of the first liquid tube 23 and the second liquid tube 24 was formed in parallel, and it formed each with the copper pipe with a smooth inner surface 0.57 mm in inside diameter, and 1.2 m in length. While the first liquid tube 23 and second liquid tube 24 supply the refrigerant liquefied with the condenser 4 to the second expansion mechanism 20, respectively, While shortening length required for the heat exchange of the first liquid tube 23 and the second liquid tube 24, and the second suction pipe 21 by this which is what carries out heat exchange to the second suction pipe 21 in the second heat exchanging part 22, with a refrigerant flow rate secured, Passage resistance can be reduced and the supercooling of a refrigerant can be secured by a thinner tube diameter. As a result, when using the inflammable refrigerant of R600a or R290 grade, the amount of liquid cooling intermediation held inside can be reduced.
[0076]Although the first liquid tube 23 and second liquid tube 24 were 0.57 mm in inside diameter, and 1.2 m in length in this embodiment, the same effect is expectable if it is two or more liquid tubes 0.5 mm or more in inside diameter. The same effect is acquired also in the cycle for cooling of the cold storage 1.
[0077]As mentioned above, while being able to do quenching efficiently by controlling resistance of the second expansion mechanism to become resistance and the equivalent grade of the first expansion mechanism at the time of overloads, such as a power up, By forming with two or more liquid tubes 0.5 mm or more in inside diameter, the liquid tube for refrigeration cycles, or the liquid tube for refrigerating cycles. While shortening length required for heat exchange, holding down very much the volume of the refrigerant which circulates the inside of a pipe to a small quantity, it is stabilized and control of flow of the first expansion mechanism and the second expansion mechanism can be performed.
[0078](Embodiment 6) Embodiment 6 by this invention is described, referring to drawings. About Embodiment 3, an identical configuration, and operation, identical codes are attached and detailed explanation is omitted.
[0079]Drawing 7 is a schematic diagram of the cooling cycle by the embodiment of the invention 6, and a refrigerator.
[0080]As shown in drawing 7, it is the second isolation wall for the first isolation wall for 25 to isolate with the first expansion valve and for 26 isolate the first expansion valve 25 with the air of the cold storage 1 and 27 to isolate with the second expansion valve, and for 28 isolate the second expansion valve 27 with the air of the freezer compartment 2. Although not illustrated, the first isolation wall 26 and second isolation wall 28 comprise flame retardant resin, and have structure which can open and close a part so that it can exchange, when an expansion valve is damaged. A setting position is near [hidden from the interior of a room] the first evaporator 5 and the second evaporator 6, and does not serve as resistance of the indoor air which carries out heat exchange to each evaporator, but is a position which can defrost an outline by the defrosting heater of the evaporator which is not illustrated.
[0081]There is the feature of the composition in this embodiment in controlling contact with the air of the refrigerator interior of a room by enclosing the first expansion valve 25 and second expansion valve 27 with the first isolation wall 26 and second isolation wall 28, respectively. While controlling heat-receiving from the outside and stabilizing flow control of the first expansion valve 25 and the second expansion valve 27 by this, when disclosure arises from a joined part etc., the adverse effect to foodstuffs can be reduced. When the inflammable refrigerant of R600a or R390 grade is used especially, the disclosure to the refrigerator interior of a room is controlled, and it is effective in the ability to reduce the danger of ignition.
[0082]In this embodiment, although installed in the interior of a room of a refrigerator, since it can avoid heat-receiving to an expansion valve if the insulation efficiency of the expansion valve of an isolation wall is enough, it may be installed in outdoor.
[0083]As mentioned above, while being able to do quenching efficiently by controlling resistance of the second expansion mechanism to become resistance and the equivalent grade of the first expansion mechanism at the time of overloads, such as a power up, the adverse effect to the foodstuffs at the time of disclosure, etc. can be inhibited by enclosing an expansion valve with an isolation wall.
[0084](Embodiment 7) Embodiment 7 by this invention is described, referring to drawings. About Embodiment 1, an identical configuration, and operation, identical codes are attached and detailed explanation is omitted.
[0085]Drawing 8 is a schematic diagram of the cooling cycle by the embodiment of the invention 7, and a refrigerator.
[0086]As shown in drawing 8, it is the third capillary, an inner diameter is a 2310-mm-long capillary [in 0.77 mm], and 29 is carrying out heat exchange to the second suction pipe 21 by the second heat exchanging part 22. The inner diameter of the first capillary 7 is a 2310-mm-long capillary [length is 2310 mm and / the second capillary 8 / an inner diameter / in 0.56 mm] in 0.77 mm. 30 is a multiple-directions selector valve which changes a refrigerant passage to the first capillary 7, second capillary 8, or third capillary 29, and the number of rotations of the compressor 3 is good transformation of 28rps to 75rps.
[0087]About the refrigerator constituted as mentioned above, the operation is explained below.
[0088]Usually, the state of each part at the time of cooling and overloads, such as a power up, is shown for comparing a conventional example with this embodiment (Table 2).
[0089]

[Table 2]

圧縮機気筒容積:5.7mL 圧縮機吸入温度:30℃			キャピラリ仕様 (内径[mm]×長さ[mm])	蒸発温度	凝縮温度	圧縮機の回転数	体積効率	循環量
				℃	℃	rps	%	kg/h
実施の形態3	通常負荷	冷蔵	φ0.77×2310	−15	30	28	80	3.1
		冷凍	φ0.56×2310	−30	30	28	70	1.4
	電源投入	冷蔵	φ0.77×2310	−27	45	75	65	4.0
		冷凍	φ0.77×2310	−27	45	75	65	4.0
従来例	通常負荷	冷蔵	φ0.77×2310	−15	30	28	80	3.1
		冷凍	φ0.56×2310	−30	30	28	70	1.4
	電源投入	冷蔵	φ0.77×2310	−27	45	75	65	4.0
		冷凍	φ0.56×2310	−38	45	75	50	1.8

[0090]At the time of the usual low loading, the number of rotations of the compressor 3 is operated to the minimum 28rps as (Table 2), The evaporating temperature of the second evaporator 6 with which inspired gas density becomes small by circulating a refrigerant to the second capillary 8 with stronger resistance than the first capillary 7 at the time of cooling of the freezer compartment 2 −30 **, An energy-saving cycle equivalent to the former is maintained by making evaporating temperature of the first evaporator 5 of the cold with large inspired gas density into −15 **.
[0091]And the compressor 3 is operated by 75rps of a maximum engine speed at the time of an overload like the power up in which load increases rapidly, Cooling can be done more quickly than before by making the amount of refrigerant circulation increase to the third capillary 29 of the time of cold storage 1 cooling and the resistance to the time of freezer compartment 2 cooling which is the amount of low refrigerant circulation by circulating a refrigerant, and acquiring high refrigerating capacity. Since the evaporating temperature of the first evaporator 5 and the second evaporator 6 acquires the heat exchanging capacity corresponding to the refrigerant flow rate which increased by considering it as −27 ** at this time, the maximum refrigerating capacity of a cooling system can be used and pulldown time is made to the shortest.
[0092]Since the capillary which is small volume is substituted for a liquid tube, can reduce refrigerant filling quantity and it is economical, and the danger of ignition at the time of revealing using prevention and the inflammable refrigerant of the adverse effect to the foodstuffs at the time of refrigerant disclosure compared with an expansion valve etc. cheaply can be reduced.
[0093]Like a power up, at the time of the heavy load which load increases by outside-air-temperature rise, when high outside air temperature is detected using an outside air temperature sensor etc., the same effect is acquired by changing to the third capillary 29.
[0094]If it is more than it, it is still wide range, and although the number of two or more capillaries is two in this embodiment, since control of flow is possible, the above effect will be acquired similarly. It may install in the cooling cycle side of the cold storage 1.
[0095]Although the channel is changed from two or more capillaries to one capillary by the multiple-directions selector valve 30 in this invention, the effect that the composition which installs an opening and closing valve before and after capillaries other than the capillary of the maximum resistance, and is opened and closed if needed is also the same is realizable.
[0096]Although the refrigerant is circulated from **** to one side if needed at the time of cooling of the freezer compartment 2, the second capillary 8 and third capillary 29 from which resistance difference is different in this embodiment, The same effect is acquired, even if simultaneous [both] or it gives only one of the two control of flow by circulating a refrigerant if needed, using the capillary of the same resistance two, The same effect will be acquired, if flow variable control can be performed so that a predetermined flow can be passed if needed by performing a circulation change using other capillary plurality.
[0097](Embodiment 8) Embodiment 8 by this invention is described, referring to drawings. About the former, an identical configuration, and operation, identical codes are attached and detailed explanation is omitted.
[0098]Drawing 9 is a schematic diagram of the cooling cycle by the embodiment of the invention 8, and air course composition.
[0099]The third heat exchanging part to which 32 considers it as the third suction pipe by 31 considering it as the third evaporator, and the first capillary 7 and second capillary 8 carry out heat exchange of 33 to the third suction pipe 32 in drawing 9, A fan for 34 to make the cold storage 1 or the freezer compartment 2 circulate through the air after the third evaporator 31 and heat exchange, The cold storage discharge duct which 35 opens the freezer compartment 2 and the cold storage 1 for free passage, and carries out the regurgitation of the air of the freezer compartment 2 to the cold storage 1, 36 — the air after the third evaporator 31 and heat exchange — the freezer compartment 2 — **** — him — a ******** discharge duct. The cold storage suction duct in which 37 leads the air in the cold storage 1 to the third evaporator 31, The freezer compartment suction duct in which 38 leads the air in the freezer compartment 2 to the third evaporator 31, and 39 are dampers which change an air course for low temperature air after the third evaporator 31 and heat exchange to the cold storage discharge duct 35 or the freezer compartment discharge duct 36, and the arrow shows the draft direction.
[0100]Although not illustrated, the discharged air temperature sensor which detects the discharged air temperature after the third evaporator 31 and heat exchange is formed in the neighborhood which the freezer compartment discharge duct 36 and the freezer compartment 2 open for free passage.
[0101]About the refrigerator constituted as mentioned above, the operation is explained below.
[0102]The simultaneous cooling mode in which the feature of the composition in this embodiment usually cools the freezer compartment 2 and the cold storage 1 simultaneously using the first capillary 7 at the time of operation, The freezing room cooling mode which cools only the freezer compartment 2 using the second capillary 8 is changed, and it cools, and is in performing operation of simultaneous cooling mode and freezing room cooling mode only using the first capillary 7 in a power up.
[0103]Usually, at the time of operation, it is controlled so that a refrigerant circulates to the first small capillary 7 of resistance by the flow path selector valve 9 first, The air which the damper 39 opened and carried out heat exchange to the third evaporator 31 by the operation of the fan 34 circulates so that it may breathe out in the freezer compartment 2 through the freezer compartment discharge duct 36, and may be breathed out by cold storage mainly through the cold storage discharge duct 35 and it may ventilate to the third evaporator 31 through the cold storage suction duct 37. This becomes the simultaneous cooling mode which cools the freezer compartment 2 and the cold storage 1 simultaneously. At this time, the number of rotations of the compressor 3 is adjusted so that evaporating temperature may turn into about −22 **.
[0104]Next, it is controlled so that a refrigerant circulates to the second large capillary 8 of resistance by the flow path selector

valve 9, The air which the damper 39 closed and carried out heat exchange to the third evaporator 31 by the operation of the fan 34 is breathed out from the freezer compartment discharge duct 36 to the freezer compartment 2, and it circulates so that heat exchange may be carried out to the third evaporator 31 through the freezer compartment suction duct 38. This becomes the freezing room cooling mode which cools only the freezer compartment 2. At this time, the number of rotations of the compressor 3 is adjusted so that evaporating temperature may turn into about -30 **.
[0105]Hereafter, it cools, changing simultaneous cooling mode and freezing room cooling mode by turns. At this time, if the cold storage 1 becomes a predetermined temperature, operation of simultaneous cooling mode will be stopped, and if the freezer compartment 2 becomes a predetermined temperature, operation of freezing room cooling mode will also be stopped.
[0106]In a power up, while operating the compressor 3 with a maximum engine speed, cooling down of simultaneous cooling mode and freezing room cooling mode is performed by turns using the first small capillary 7 of resistance. At this time, the number of rotations of the compressor 3 is adjusted so that evaporating temperature may turn into about -27 **. And if the freezer compartment 2 becomes a predetermined temperature, while stopping cooling down, it usually changes to control of operation.
[0107]As a result, in the simultaneous cooling mode at the time of operation, since the air temperature which carries out heat exchange compared with freezing room cooling mode is high, cooling with high evaporating temperature with high theoretical efficiency is attained, and synthetic cooling efficiency can usually be improved. Although evaporating temperature becomes low compared with cooling mode independent [cold storage 1], there is an advantage which can set up cooling operation time for a long time. This has a low air temperature which carries out heat exchange compared with cooling mode independent [cold storage 1], and since the freezer compartment 2 may be warmed in the evaporating temperature more than the air temperature of the freezer compartment 2, the evaporating temperature of simultaneous cooling mode is because -20 ** order becomes a limit.
[0108]In a power up, since the cold storage 1 and the freezer compartment 2 are cooled using the maximum capacity of a cooling system, pulldown time from after powering on can be made into the shortest.
[0109]At the time of overloads, such as operation after defrosting the third evaporator 31, even if it performs freezing room cooling mode using the first capillary 7 like a power up, the effect which quenches the freezer compartment 2 is acquired. If the number of rotations of the compressor 3 is made to increase and evaporating temperature is maintained while detecting the rise of the discharged air temperature after the third evaporator 31 and heat exchange and changing to the first capillary 7 when loads, such as a foodstuffs injection, increase rapidly in freezing room cooling mode, the effect which quenches the freezer compartment 2 similarly will be acquired.
[0110]As mentioned above, while being able to do quenching efficiently by performing cooling down of simultaneous cooling mode and freezing room cooling mode by turns using the first small capillary of resistance at the time of overloads, such as a power up, By cooling cold storage by simultaneous cooling mode, cold storage operation time at the time of operation can usually be lengthened, and the temperature change of cold storage can be controlled.
[0111](Embodiment 9) Embodiment 9 by this invention is described, referring to drawings. About the same composition and operation as Embodiment 3, identical codes are attached and detailed explanation is omitted.
[0112]The cooling cycle and refrigerator by the embodiment of the invention 9 are the same as that of Embodiment 1 shown by drawing 1. Drawing 10 is a figure showing the evaporating temperature of the first evaporator 5 and the second evaporator 6, and the relation of evaporating capacity.
[0113]In drawing 10, the evaporating capacity of the first evaporator 5 is predetermined evaporating temperature, and shows the refrigerant flow rate which can carry out heat exchange to the air of the cold storage 1, and can be evaporated. Similarly, the evaporating capacity of the second evaporator 6 is predetermined evaporating temperature, and shows the refrigerant flow rate which can carry out heat exchange to the air of the freezer compartment 2, and can be evaporated. It is largely based on the difference of the air temperature which carries out heat exchange that the evaporating capacity of the first evaporator 5 and the evaporating capacity of the second evaporator 6 have a big difference. Therefore, when there is no difference with it, the evaporating capacity of the first evaporator 5 and the second evaporator 6 is almost equivalent, and higher than the evaporating capacity of the first evaporator 5 shown in drawing 10. [a high air temperature which carries out heat exchange like a power up, and] [big]
[0114]The operation at the time of usual operation of this embodiment is explained below.
[0115]A refrigerant flow rate required for the cooling system corresponding to the endothermic load of the refrigerator at predetermined outdoor air temperature is set up, and the control table which specified outdoor air temperature and the relation of the refrigerant flow rate beforehand is set up. Usually, at the time of operation, the refrigerant flow rate made into a target is determined from the outdoor air temperature detected with the outdoor air temperature sensor (not shown), and said control table.
[0116]Here, since the endothermic load of the refrigerator at predetermined outdoor air temperature is controllable by the operating condition it is more efficient to assume the quantity of heat which flows through the thermal insulation 13 of the refrigerator body 12 which contains neither door opening closed load nor the load of a foodstuffs injection, it is desirable. If it sets up to such an extent that the refrigerant flow rate specified beforehand can cool predetermined endothermic load with 70 to 80% of operating efficiency, it can respond change factors, such as door opening closed load and a foodstuffs injection, to some extent by the increase in operating efficiency comparatively efficiently. Endothermic load has very small outdoor air temperature below 10 **, and on efficiency, in the case which is not preferred, feebleminded power-ization of a cooling system may set up a refrigerant flow rate so that operating efficiency may become low.
[0117]Next, the resistance of the expansion mechanism 16 and the expansion mechanism 20 and the capability of the condenser 4 are adjusted so that it may become a target refrigerant flow rate. While adjusting the resistance of the expansion mechanism 16 and the expansion mechanism 20 supposing the refrigerant state of the entrance of the expansion mechanism 16 or the expansion mechanism 20 becoming about 0 ** of supercooling at this time, it is desirable on cycle efficiency to adjust the capability of the condenser 4 so that it may not have a big dryness fraction.
[0118]And in a target refrigerant flow rate, the number of rotations of the compressor 3 is adjusted so that the first evaporator 5 and second evaporator 6 may become the evaporating temperature which shows the maximum capacity. At this embodiment, the first evaporator 5 and second evaporator 6 operate in the state which showed by the A point and B point of drawing 8. here — the cold storage 1 and the freezer compartment 2 — the outside-air-temperature dependency of endothermic load — things — a thing, and the first evaporator 5 and second evaporator 6 — the maximum capacity — the cold storage 1 since it is greatly different, and the freezer compartment 2 — it is desirable to adjust the number of rotations of the compressor 3 independently, respectively.

[0119]When operating efficiency reaches to about 100% exceeding prediction, the change factor of endothermic loads, such as door opening closed load and a foodstuffs injection, carries out the increase in the specified quantity of the desired value of the amount of refrigerant circulation specified with said control table, and should just perform same control. At this time, from change of outlet air temperature by which heat exchange was carried out to the first evaporator 5 or the second evaporator 6, the rapid increase in endothermic load may be detected and the increase in the specified quantity of the desired value of the amount of refrigerant circulation may be carried out.
[0120]As a result, the evaporating temperature of the first evaporator 5 and the second evaporator 6 set up according to the endothermic burden at the outdoor air temperature of 25 ** which is anticipated-use conditions, For example, by fluctuating evaporating temperature according to endothermic load compared with the cooling system by which operation control was carried out fixed at -15 ** and -30 **, when especially endothermic load is small, theoretical efficiency can be raised to the maximum, and the amount of used electricity of **** of a refrigerator can be reduced. The effect which reduces amount of used electricity especially in the refrigerator of high heat insulation performance using the change system which controls independently the cold storage 1 which differs in the outside-air-temperature dependency of endothermic load, and the freezer compartment 2 is large.
[0121]Although resistance controlled the refrigerant flow rate in this embodiment using the expansion mechanism 16 and the expansion mechanism 20 which are changed arbitrarily, Fixed resistance of the capillary etc. from which a refrigerant flow rate changes appropriately to outdoor air temperature, i.e., condensation temperature, may be used, several capillaries from which resistance differs may be changed, and a refrigerant flow rate may be controlled.
[0122](Embodiment 10) Embodiment 10 by this invention is described, referring to drawings. About Embodiment 1 and Embodiment 7, an identical configuration, and an operation, identical codes are attached and detailed explanation is omitted.
[0123]Drawing 11 is a schematic diagram of the cooling cycle by the embodiment of the invention 10, and a refrigerator, and drawing 12 is a sectional view of a receiver, and a schematic diagram of a refrigerator system.
[0124]It is the receiver with which 41 was provided between the condenser 4 and the flow path selector valve 9 in drawing 11 and drawing 12.
[0125]About the refrigerator constituted as mentioned above, the operation is explained below.
[0126]Usually, when changing from cooling of the freezer compartment 2 at the time to cooling of the cold storage 1, it extracts from the second expansion mechanism 20, and shifts to the cycle of the first expansion mechanism 16 with a small quantity. At this time, the liquid cooling intermediation which was stagnating in the receiver 41 flows into the first expansion mechanism through the liquid tube 15 for refrigeration cycles, the amount of refrigerant circulation increases, and it is stabilized in the predetermined amount of high refrigerant circulation at an early stage.
[0127]And in a power up, the compressor 3 is operated with a maximum engine speed, and resistance of the second expansion mechanism 20 is controlled to become resistance and the equivalent grade of the first expansion mechanism 16. as a result, the increase in until comparable of the refrigerant flow rate at the time of freezer compartment 2 cooling is carried out with the refrigerant flow rate at the time of cold storage 1 cooling, and it quenches efficiently by making it the evaporating temperature which acquires the heat exchanging capacity corresponding to a refrigerant flow rate.
[0128]As mentioned above, can quench efficiently at the time of overloads, such as a power up, and. Usually, since a refrigerant required for the predetermined amount of high refrigerant circulation at the time of cooling of the cold storage 1 flows from the receiver 41, is stabilized in the predetermined amount of high refrigerant circulation at an early stage and shifts to the low compression ratio state which is the fitness of compressor efficiency at the time of the change to cooling of the cold storage 1 from cooling of the freezer compartment 2 in load, the power consumption of a compressor decreases.
[0129]
[Effect of the Invention]As explained above, the invention of this invention according to claim 1, The refrigerant circuit which is the refrigerator provided with the refrigeration field and the refrigerating area in the adiabatic box, has an evaporator in said refrigeration field and said refrigerating area, respectively, and pours a refrigerant to the evaporator of said refrigeration field at least, In what provides the refrigerant circuit which pours a refrigerant to the evaporator of said refrigerating area, changes these refrigerant circuits, and is cooled, Since the endothermic burden of said refrigerating area at the time of the stable operation on the standard cooling conditions of a refrigerator was made into the endothermic burden of said refrigeration field below equivalent, Since being able to maintain the cooling operation time of the refrigeration field where refrigerating capacity is large, for example, becoming the extreme low operating efficiency of 15% or less by controlling the cooling operation time of the refrigerating area where refrigerating capacity is comparatively low can prevent, While the temperature control of a refrigeration field becomes easy, the rate of the cooling loss at the time of compressor start is controlled, and operation efficient as a result can be attained.
[0130]It can maintain the cooling operation time of the refrigeration field where refrigerating capacity is large while it secures effective content volume, without thickening an adiabatic wall, since the adiabatic wall of the adiabatic box was formed with the foamed heat insulating material and the invention according to claim 2 allocated the vacuum insulation material in said adiabatic wall of a refrigerating area.
[0131]Since the adiabatic wall of the adiabatic box was formed with the foamed heat insulating material and the invention according to claim 3 allocated the vacuum insulation material in said adiabatic wall in 50 to 80% of range of outer packaging surface area, While securing effective content volume, without thickening an adiabatic wall, high cost performance is obtained by allocating a vacuum insulation material effectively.
[0132]The invention according to claim 4 is the refrigerator provided with the refrigeration field and the refrigerating area in the adiabatic box, and have the first evaporator to said refrigeration field, have the second evaporator in said refrigerating area, and A compressor, While constituting a condenser, a flow path selector valve, the liquid tube for refrigeration cycles, said first evaporator, and the first suction pipe that carries out heat exchange to said liquid tube for refrigeration cycles from a closed loop, So that it may become in parallel with said liquid tube for refrigeration cycles, said first expansion mechanism, said first evaporator, and said first suction pipe The liquid tube for refrigerating cycles, The second expansion mechanism, said second evaporator, and the second suction pipe that carries out heat exchange to said liquid tube for refrigerating cycles, It is what performs mutually cooling of said refrigeration field and said refrigerating area independently by connecting a check valve and changing the flow of a refrigerant by said flow path selector valve, Since it is a refrigerator, wherein a power up makes resistance of said second expansion mechanism smaller than the resistance at the time of the stable operation on the standard cooling conditions of a refrigerator, A cooling state can be promptly stabilized as an amount of high refrigerant circulation equivalent to the time of refrigeration field cooling at the time of refrigerating area cooling at the time of overload operation, such as a power

up.
[0133]Since the liquid tube for refrigeration cycles and the liquid tube for refrigerating cycles are characterized by an inside diameter being 0.8 mm or more, the invention according to claim 5, At the time of overload operation, such as a power up, the volume of the refrigerant which stagnates in the liquid tube for refrigeration cycles or the liquid tube for refrigerating cycles while carrying out the amount of high refrigerant circulation and promoting quick cooling equivalent to the time of refrigeration field cooling is controlled in a small quantity, it is stabilized at the time of refrigerating area cooling, and it can perform control of flow of an expansion mechanism.
[0134]The invention according to claim 6 is formed with two or more liquid tubes with which the liquid tube for refrigeration cycles or the liquid tube for refrigerating cycles was parallel, and since said liquid tube is 0.5 mm or more, an inside diameter, The heat exchange length of a suction pipe and a liquid tube is shortened, the volume of the refrigerant which stagnates in the liquid tube for refrigeration cycles or the liquid tube for refrigerating cycles is controlled in a small quantity, it is stabilized and control of flow of an expansion mechanism can be performed.
[0135]Since the first expansion mechanism and second expansion mechanism are the expansion valve installed in the air in a warehouse, and the isolated portion, the invention according to claim 7 can control that a refrigerant is revealed to the interior of a room at the time of refrigerant disclosure.
[0136]The invention according to claim 8 forms the first expansion mechanism or second expansion mechanism by two or more capillaries which carry out heat exchange to the first suction pipe or the second suction pipe, Since resistance is changed by substituting said two or more capillaries for the liquid tube for refrigeration cycles, or the liquid tube for refrigerating cycles, and changing the channel of two or more capillaries, Maintaining energy saving due to a raise in evaporating temperature at the time of refrigeration field cooling, and the fall of the entrance refrigerant dryness of the expansion mechanism for refrigerating area cooling at the time of the stable operation on the standard cooling conditions of a refrigerator. In addition to the ability to do quenching for a power up efficiently, refrigerant filling quantity can be reduced by substituting for a liquid tube the capillary which is small volume.
[0137]The invention according to claim 9 is a refrigeration field and a refrigerating area the refrigerator which it had, and A compressor, A condenser, a flow path selector valve, the first capillary, and the second capillary, The third evaporator and the third suction pipe that carries out heat exchange to said first capillary and the second capillary, The first air course that carries out heat exchange of the air in said refrigeration field to said third evaporator, Have the second air course that carries out heat exchange of the air in a refrigerating area to said third evaporator, and constitute said compressor, said condenser, said flow path selector valve, said first capillary, said third evaporator, and said third suction pipe from a closed loop, and. By connecting said second capillary so that it may become said first capillary and parallel, and changing the flow of the refrigerant to a capillary by said flow path selector valve, Since it is a refrigerator changing the flow of a refrigerant using said second capillary when using said first capillary when opening said first air course and said second air course, and opening only the second air course, Can quench efficiently by performing by turns cooling down which cools a refrigeration field and a refrigerating area simultaneously by the first small capillary of resistance at the time of overloads, such as a power up, and cooling down which cools only a refrigerating area, and. At the time of the stable operation on the standard cooling conditions of a refrigerator, refrigeration field cool time is lengthened by cooling a refrigeration field and a refrigerating area simultaneously, and the temperature change in a refrigeration field can be controlled.
[0138]The compressor of the invention according to claim 10 is number-of-rotations good transformation, and, as for the first expansion mechanism and second expansion mechanism, the amount of diaphragms can change, Have an outside air temperature sensor which detects outdoor air temperature, and the amount of diaphragms of said first expansion mechanism and the second expansion mechanism is controlled so that the required refrigerant flow rate equivalent to the burden computed from the outside air temperature which said outside air temperature sensor detected circulates, From claim 4 controlling the number of rotations of said compressor to become predetermined evaporating temperature from said required refrigerant flow rate, since it is a refrigerator of nine given in any 1 paragraph, At the time of overloads, such as a power up, it can quench efficiently, and it becomes the evaporating temperature which can always acquire the heat exchanging capacity corresponding to a refrigerant flow rate, and cools efficiently using the maximum capacity of a refrigerating cycle.
[0139]Since the invention according to claim 11 is a refrigerator of ten given in any 1 paragraph from claim 4 which formed the receiver between the condenser and the flow path selector valve, A temporary shortage of the amount of refrigerant circulation when changing from refrigerating area cooling of the amount of low refrigerant circulation to refrigeration field cooling of the amount of high refrigerant circulation is relieved, and it can shift to the efficient cycle of cold storage cooling at an early stage.

[Translation done.]

DESCRIPTION OF DRAWINGS

[Brief Description of the Drawings]
[Drawing 1]The cooling cycle in the embodiment of the invention 1, and the schematic diagram of a refrigerator
[Drawing 2]The cooling cycle in the embodiment of the invention 2, and the schematic diagram of a refrigerator
[Drawing 3]The cooling cycle in the embodiment of the invention 3, and the schematic diagram of a refrigerator
[Drawing 4]The P-h diagram of the cooling cycle in the embodiment of the invention 4
[Drawing 5]The cooling cycle in the embodiment of the invention 5, and the schematic diagram of a refrigerator
[Drawing 6]The strabism sectional view of the important section in the embodiment of the invention 5
[Drawing 7]The cooling cycle in the embodiment of the invention 6, and the schematic diagram of a refrigerator
[Drawing 8]The cooling cycle in the embodiment of the invention 7, and the schematic diagram of a refrigerator
[Drawing 9]The cooling cycle in the embodiment of the invention 8, and the schematic diagram of air course composition
[Drawing 10]The characteristic figure of the evaporating temperature of an evaporator, and evaporating capacity in the embodiment of the invention 9
[Drawing 11]The cooling cycle in the embodiment of the invention 10, and the schematic diagram of a refrigerator
[Drawing 12]The sectional view of the receiver in the embodiment of the invention 10, and the schematic diagram of a refrigerator system
[Drawing 13]The cooling cycle of the conventional refrigerator, and the schematic diagram of a refrigerator
[Drawing 14]The refrigerant flow rate characteristic figure of the expansion mechanism of the conventional refrigerator
[Description of Notations]
1 Cold storage
2 Freezer compartment
3 Compressor
4 Condenser
5 The first evaporator
6 The second evaporator
7 The first capillary
8 The second capillary
9 Flow path selector valve
13 Thermal insulation
14 Check valve
15 The liquid tube for refrigeration cycles
16 The first expansion mechanism
17 The first suction pipe
19 The liquid tube for refrigerating cycles
20 The second expansion mechanism
21 The second suction pipe
31 The third evaporator
32 The third suction pipe
40 Vacuum insulation material
41 Receiver

[Translation done.]

(19)日本国特許庁（ＪＰ）　(12) 公開特許公報（Ａ）　(11)特許出願公開番号
特開2002－372319
（P2002－372319A）
(43)公開日　平成14年12月26日(2002. 12. 26)

(51)Int.Cl.⁷		識別記号	ＦＩ			テーマコード（参考）
Ｆ２５Ｂ	5/02	５３０	Ｆ２５Ｂ	5/02	５３０Ｈ	３Ｌ０４５
	1/00	３３１		1/00	３３１Ｚ	３Ｌ１０２
Ｆ２５Ｄ	11/00	１０１	Ｆ２５Ｄ	11/00	１０１Ｂ	
	23/06			23/06	Ｘ	

審査請求　未請求　請求項の数11　ＯＬ　（全 17 頁）

(21)出願番号　特願2001－184513(P2001－184513)

(22)出願日　平成13年６月19日(2001. 6. 19)

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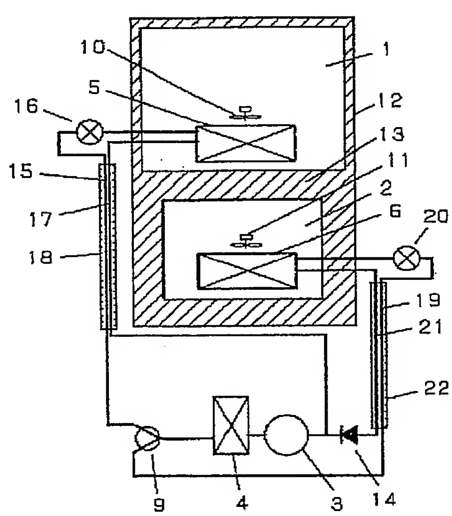
最終頁に続く

(54)【発明の名称】　冷蔵庫

(57)【要約】
【課題】　冷蔵庫の冷却運転時に安定した温度制御を効率よく行える断熱箱体の吸熱負荷構成を提供する。
【解決手段】　冷蔵室１と冷凍室２にそれぞれ第一の蒸発器５，第二の蒸発器６を有し、第一の蒸発器５の冷媒回路と第二の蒸発器６の冷媒回路を流路切替弁９で切り替えて冷却し、冷蔵庫の標準的な冷却条件における安定運転時の冷凍室２の吸熱負荷量を冷蔵室１の吸熱負荷量と同等以下にしたので、比較的冷凍能力が低い冷凍室２の冷却運転時間を抑制することで、冷凍能力が大きい冷蔵室１の冷却運転時間を維持することができ、極端な低運転率になることが防止できるので、冷蔵室１の温度制御が容易になるとともに、圧縮機３の起動時の冷却ロスを抑制し効率的な運転ができる。

１　冷蔵室（冷蔵領域）
２　冷凍室（冷凍領域）
３　圧縮機
４　凝縮器
５　第一の蒸発器
６　第二の蒸発器
９　流路切替弁
１３　断熱材
１４　逆止弁
１５　冷蔵サイクル用液管
１６　第一の膨張機構
１７　第一の吸入管
１９　冷凍サイクル用液管
２０　第二の膨張機構
２１　第二の吸入管



【特許請求の範囲】

【請求項１】　断熱箱体内に冷蔵領域と冷凍領域を備えた冷蔵庫であって、前記冷蔵領域と前記冷凍領域にそれぞれ蒸発器を有し、少なくとも前記冷蔵領域の蒸発器に冷媒を流す冷媒回路と、前記冷凍領域の蒸発器に冷媒を流す冷媒回路とを設けてこれら冷媒回路を切り替えて冷却するものにおいて、冷蔵庫の標準的な冷却条件における安定運転時の前記冷凍領域の吸熱負荷量を前記冷蔵領域の吸熱負荷量と同等以下にしたことを特徴とする冷蔵庫。

【請求項２】　断熱箱体の断熱壁は発泡断熱材で形成され、冷凍領域の前記断熱壁には真空断熱材を配設したことを特徴とする請求項１に記載の冷蔵庫。

【請求項３】　断熱箱体の断熱壁は発泡断熱材で形成され、前記断熱壁には外箱表面積の５０～８０％の範囲で真空断熱材を配設したことを特徴とする請求項１に記載の冷蔵庫。

【請求項４】　断熱箱体内に冷蔵領域と冷凍領域を備えた冷蔵庫であって、前記冷蔵領域に第一の蒸発器、前記冷凍領域に第二の蒸発器を有し、圧縮機と、凝縮器と、流路切替弁と、冷蔵サイクル用液管と、前記第一の蒸発器と、前記冷蔵サイクル用液管と熱交換する第一の吸入管とを閉ループで構成するとともに、前記冷蔵サイクル用液管と前記第一の膨張機構と前記第一の蒸発器と前記第一の吸入管とに並列になるように冷凍サイクル用液管と、第二の膨張機構と、前記第二の蒸発器と、前記冷凍サイクル用液管と熱交換する第二の吸入管と、逆止弁とを接続し、前記流路切替弁により冷媒の流れを切り替えることで前記冷蔵領域と前記冷凍領域の冷却を互いに独立して行うものであり、電源投入時は前記第二の膨張機構の抵抗を冷蔵庫の標準的な冷却条件における安定運転時の抵抗より小さくすることを特徴とする冷蔵庫。

【請求項５】　冷蔵サイクル用液管および冷凍サイクル用液管は内径が０．８mm以上であることを特徴とする請求項４に記載の冷蔵庫。

【請求項６】　冷蔵サイクル用液管あるいは冷凍サイクル用液管は並行した複数の液管で形成され、前記液管は内径が０．５mm以上であることを特徴とする請求項４に記載の冷蔵庫。

【請求項７】　第一の膨張機構と第二の膨張機構は庫内空気と隔離された部分に設置した膨張弁であることを特徴とする請求項４から６のいずれか一項に記載の冷蔵庫。

【請求項８】　第一の膨張機構あるいは第二の膨張機構を第一の吸入管あるいは第二の吸入管と熱交換する複数のキャピラリで形成し、冷蔵サイクル用液管あるいは冷凍サイクル用液管を複数の前記キャピラリで代用し、複数のキャピラリの流路を切り替えることで抵抗を変化させることを特徴とする請求項４に記載の冷蔵庫。

【請求項９】　冷蔵領域と冷凍領域を備えた冷蔵庫であって、圧縮機と、凝縮器と、流路切替弁と、第一のキャピラリと、第二のキャピラリと、第三の蒸発器と、前記第一のキャピラリ及び第二のキャピラリと熱交換する第三の吸入管と、前記第三の蒸発器と前記冷蔵領域内の空気を熱交換する第一の風路と、前記第三の蒸発器と冷凍領域内の空気を熱交換する第二の風路とを備え、前記圧縮機と前記凝縮器と前記流路切替弁と前記第一のキャピラリと前記第三の蒸発器と前記第三の吸入管とを閉ループで構成すると共に、前記第一のキャピラリと並列になるように前記第二のキャピラリとを接続し、前記流路切替弁によりキャピラリへの冷媒の流れを切り替えることにより、前記第一の風路と前記第二の風路を開く時は前記第一のキャピラリを使用し、第二の風路のみを開く時は前記第二のキャピラリを使用して冷媒の流量を可変することを特徴とする冷蔵庫。

【請求項１０】　圧縮機は回転数可変型であり、第一の膨張機構と第二の膨張機構は絞り量が変化可能であり、外気温度を検知する外気温センサを有し、前記第一の膨張機構と第二の膨張機構の絞り量は前記外気温センサが検知した外気温から算出した負荷量に相当する必要冷媒流量が流通するように制御され、前記圧縮機の回転数は前記必要冷媒流量から所定蒸発温度になるように制御することを特徴とする請求項４から９のいずれか一項記載の冷蔵庫。

【請求項１１】　凝縮器と流路切替弁の間に受液器を設けた請求項４から１０のいずれか一項記載の冷蔵庫。

【発明の詳細な説明】

【０００１】

【発明の属する技術分野】本発明は冷蔵室と冷凍室を別々の蒸発器で独立して冷却することで高効率化を図った冷蔵庫に関するものである。

【０００２】

【従来の技術】近年、冷蔵室と冷凍室を別々の蒸発器を有する冷蔵庫に関するものとしては、図１３に従来の冷却サイクル並びに冷蔵庫の一例として、特開平１１－１４８７６１号公報に開示されている冷蔵庫の概略図を示す。

【０００３】１は冷蔵室、２は冷凍室、３は圧縮機、４は凝縮器、５は冷蔵室１内に配設された第一の蒸発器であり、６は冷凍室２内に配設された第二の蒸発器である。

【０００４】７は冷蔵室冷却用である第一の蒸発器５の冷媒回路上流側に配設された第一のキャピラリであり、８は冷凍室冷却用である第二の蒸発器６の冷媒回路上流側に配設された第二のキャピラリであり、９は冷媒の流路を切り替える流路切替弁、１０は第一の蒸発器５と熱交換した冷気を冷蔵室１に循環させるための第一のファン、１１は第二の蒸発器６と熱交換した冷気を冷凍室２に循環させるための第二のファン、１２は冷蔵庫本体、１３は外気から室内への熱侵入を抑制する断熱材であ

る。

【０００５】以上のように構成された従来例の冷蔵庫について、以下その動作を説明する。

【０００６】冷凍サイクルの運転は以下のように行われる。まず圧縮機３により圧縮された冷媒が凝縮器４で凝縮液化される。凝縮された冷媒は第一のキャピラリ７もしくは第二のキャピラリ８で減圧されて、それぞれ第一の蒸発器５、第二の蒸発器６へ流入、蒸発気化された後、再び圧縮機３へと吸入される。

【０００７】第一のファン１０、第二のファン１１により、冷媒が蒸発気化して比較的低温となった第一の蒸発器５、第二の蒸発器６と冷蔵室１、冷凍室２の空気が熱交換して冷気が循環することで各室が冷却される。

【０００８】冷凍冷蔵庫の冷却運転は図示しない各室の温度検知手段と制御手段により以下のように行われる。

【０００９】冷蔵室１、冷凍室２の各温度検知手段が所定値以上の温度上昇を検知すると圧縮機３が起動し、所定値以下となるまで冷凍サイクルの運転が行われる。

【００１０】冷蔵室１の温度検知手段が所定値以上となった場合、流路切替弁９により冷媒は第二の蒸発器６には流入することなく、第一の蒸発器５へのみ流れる。このときの蒸発温度は冷蔵室１の温度設定が５℃程度に対して０～－１５℃であり、－２５～－３０℃の蒸発温度で運転される場合に比べて２～２．５倍の成績係数で圧縮機の運転が行われる。

【００１１】冷凍室２の温度検知手段が所定値以上となった場合、流路切替弁９により冷媒は第二の蒸発器６へと流入し、冷凍室２の冷却が行われる。このときの蒸発温度は冷凍室の温度設定が－１８℃程度に対し通常の蒸発温度－２５℃から－３０℃で冷却される。

【００１２】また、圧縮機３は電源投入時に最高回転数で運転を行い、冷蔵庫の標準的な冷却条件における安定運転時には最低回転数で運転を行っている。

【００１３】以上のように冷蔵室１と冷凍室２とを交互に繰り返し冷却するので、冷蔵室１冷却時は独立的に冷媒を第一の蒸発器へと循環させることで高蒸発温度（０～－２０℃）が可能であり、圧縮機３の圧縮比を小さくでき、高い成績係数で運転を行い効率化を図ると共に、冷蔵室１の室温と蒸発温度との差を小さくすることで温度変動を低減させて冷蔵室１の均温化を狙っている。また、圧縮機３は電源投入時に最高回転数で運転して急冷を行い、冷蔵庫の標準的な冷却条件における安定運転時は最低回転数で運転して、蒸発温度を上げることで更なる省エネルギー化を行っている。

【００１４】ここで、例えば、第一の蒸発器５の蒸発温度を－１０℃、第二の蒸発器６の蒸発温度を－３０℃とし冷媒としてＨＦＣ１３４ａを用いると、第一の蒸発器５で蒸発する冷媒ガスの密度が第二の蒸発器６で蒸発する冷媒ガスの密度の約２．３倍となる。同様に冷媒としてＨＣ６００ａを用いても約２．２倍となる。

【００１５】この結果、通常負荷時の冷蔵室冷却と冷凍室冷却の圧縮機３の回転数を同一とする場合は第一のキャピラリ７に対して第二のキャピラリ８の抵抗を約２倍に設定して第二の蒸発器６に流れる冷媒量を小さくして－３０℃の蒸発温度を実現する。また、通常負荷時の冷蔵室冷却と冷凍室冷却の圧縮機３の回転数を変化させる場合は第一のキャピラリ７と第二のキャピラリ８の抵抗をほぼ同一、すなわち冷媒流量をほぼ同一として、冷凍室冷却を行うときに回転数を上げて－３０℃の蒸発温度を実現することも可能である。

【００１６】

【発明が解決しようとする課題】しかしながら、上記従来の構成では、特に吸熱負荷の小さい高断熱性能の冷蔵庫において冷蔵領域の吸熱負荷比率が小さい場合、冷蔵室冷却サイクルの運転時間が極端に小さくなり、冷蔵室の温度制御が困難になるとともに、圧縮機起動時の冷却ロスの割合が大きくなり結果として効率的な運転ができなくなるという欠点があった。

【００１７】本発明は従来の課題を解決するもので、冷蔵庫の冷却運転時に安定した温度制御を効率よく行える断熱箱体の吸熱負荷構成を実現することを目的としている。

【００１８】また、冷蔵室冷却サイクル運転時に圧縮機の回転数を下げて対応すると、圧縮機の回転数範囲に限界があるため、冷凍室冷却サイクル運転時の能力可変範囲が限定され、結果として電源投入時や除霜復帰時のような負荷が急増した場合等の過負荷運転時における冷凍室冷却サイクルの冷凍能力が十分得られない問題が生じる。

【００１９】さらに、キャピラリの抵抗を固定すると、電源投入時や除霜復帰時のような負荷が急増した場合等の過負荷運転時において、冷媒ガス密度が小さい冷凍室冷却サイクルの冷凍能力を増加させることが困難となるという欠点を有していた。これは、省エネルギー化を目指した高断熱性能の冷蔵庫においては、冷蔵庫の標準的な冷却条件における安定運転時に必要な著しく低い冷凍能力に合わせて、圧縮機の能力やキャピラリの抵抗を最適化する方が、総合的に高い効率が得られるためである。

【００２０】例えば、図１４に示したように、比較的高い外気温の吸熱負荷量に必要な冷媒流量に合わせたキャピラリＡでは、比較的低い外気温では必要以上の冷媒流量が流れ、結果として冷媒ガスの比率、すなわち冷媒の乾き度が増加して自動的に流量調整が行われることになる。比較的低い外気温の吸熱負荷量に必要な冷媒流量に合わせたキャピラリＢでは、冷媒ガスによる調整代は小さくなるが、比較的高い外気温では冷媒量不足となるというものである。

【００２１】本発明の他の目的は、電源投入時や除霜復帰時等の過負荷運転時に効率が高く迅速な冷却機能を提

供することを目的としている。

【００２２】

【課題を解決するための手段】本発明の請求項１に記載の発明は、断熱箱体内に冷蔵領域と冷凍領域を備えた冷蔵庫であって、前記冷蔵領域と前記冷凍領域にそれぞれ蒸発器を有し、少なくとも前記冷蔵領域の蒸発器に冷媒を流す冷媒回路と、前記冷凍領域の蒸発器に冷媒を流す冷媒回路とを設けてこれら冷媒回路を切り替えて冷却するものにおいて、冷蔵庫の標準的な冷却条件における安定運転時の前記冷凍領域の吸熱負荷量を前記冷蔵領域の吸熱負荷量と同等以下にしたことを特徴とする冷蔵庫であるので、比較的冷凍能力が低い冷凍領域の冷却運転時間を抑制することで、冷凍能力が大きい冷蔵領域の冷却運転時間を維持することができ、１５％以下の極端な低運転率になることが防止できるので、冷蔵領域の温度制御が容易になるとともに、圧縮機起動時の冷却ロスの割合を抑制し結果として効率的な運転が達成できる。

【００２３】本発明の請求項２に記載の発明は、断熱箱体の断熱壁は発泡断熱材で形成され、冷凍領域の前記断熱壁には真空断熱材を配設したことを特徴とする請求項１に記載の冷蔵庫であるので、断熱壁を厚くせずに有効内容積を確保するとともに、冷凍能力が大きい冷蔵領域の冷却運転時間を維持することができる。

【００２４】本発明の請求項３に記載の発明は、断熱箱体の断熱壁は発泡断熱材で形成され、前記断熱壁には外箱表面積の５０～８０％の範囲で真空断熱材を配設したことを特徴とする請求項１に記載の冷蔵庫であるので、断熱壁を厚くせずに有効内容積を確保するとともに、効果的に真空断熱材を配設することで高いコストパフォーマンスが得られる。

【００２５】本発明の請求項４に記載の発明は、断熱箱体内に冷蔵領域と冷凍領域を備えた冷蔵庫であって、前記冷蔵領域に第一の蒸発器、前記冷凍領域に第二の蒸発器を有し、圧縮機と、凝縮器と、流路切替弁と、冷蔵サイクル用液管と、前記第一の蒸発器と、前記冷蔵サイクル用液管と熱交換する第一の吸入管とを閉ループで構成するとともに、前記冷蔵サイクル用液管と前記第一の膨張機構と前記第一の蒸発器と前記第一の吸入管とに並列になるように冷凍サイクル用液管と、第二の膨張機構と、前記第二の蒸発器と、前記冷凍サイクル用液管と熱交換する第二の吸入管と、逆止弁とを接続し、前記流路切替弁により冷媒の流れを切り替えることで前記冷蔵領域と前記冷凍領域の冷却を互いに独立して行うものであり、電源投入時は前記第二の膨張機構の抵抗を冷蔵庫の標準的な冷却条件における安定運転時の抵抗より小さくすることを特徴とする冷蔵庫であるので、冷蔵庫の標準的な冷却条件における安定運転時において従来と同じ冷蔵領域冷却時の高蒸発温度を得ると共に冷凍領域冷却時にガス冷媒の循環を低減して低負荷に対応した低冷媒流量を得ることで省エネルギーサイクルを維持しながら、電源投入時等の過負荷運転時に冷凍領域冷却時は冷蔵領域冷却時と同等の高冷媒循環量とすると共に、その冷媒循環量に対応した熱交換能力となる蒸発温度とすることで効率良く急冷を行う。

【００２６】本発明の請求項５に記載の発明は、冷蔵サイクル用液管および冷凍サイクル用液管は内径が０．８ｍｍ以上であることを特徴とする請求項４記載の冷蔵庫であるので、冷蔵庫の標準的な冷却条件における安定運転時において省エネルギー化を維持しながら、電源投入時等の過負荷運転時に冷凍領域冷却時は冷蔵領域冷却時と同等の高冷媒循環量とすると共に、その冷媒循環量に対応した熱交換能力となる蒸発温度とすることで効率良く急冷を行う。また、冷蔵サイクル用液管あるいは冷凍サイクル用液管に滞留する冷媒の液量を少量に抑制して膨張機構の流量制御を安定して行うことができる。

【００２７】本発明の請求項６に記載の発明は、冷蔵サイクル用液管あるいは冷凍サイクル用液管は並行した複数の液管で形成され、前記液管は内径が０．５ｍｍ以上であることを特徴とする請求項４記載の冷蔵庫であるので、冷蔵庫の標準的な冷却条件における安定運転時において冷蔵領域冷却時の高蒸発温度化と冷凍領域冷却用膨張機構の入口冷媒乾き度の低下により省エネルギー化を維持しながら、電源投入時に効率良く急冷ができることに加えて、吸入管と液管との熱交換長さを短くすると共に、冷蔵サイクル用液管あるいは冷凍サイクル用液管に滞留する冷媒の液量を少量に抑制して膨張機構の流量制御を安定して行うことができる。

【００２８】本発明の請求項７に記載の発明は、第一の膨張機構と第二の膨張機構は庫内空気と隔離された部分に設置した膨張弁であることを特徴とする請求項４から６のいずれか一項記載の冷蔵庫であるので、冷蔵庫の標準的な冷却条件における安定運転時において冷蔵領域冷却時の高蒸発温度化と冷凍領域冷却用膨張機構の入口冷媒乾き度の低下により省エネルギー化を維持しながら、電源投入時に効率良く急冷ができることに加えて、冷媒漏洩時に冷媒が室内へ漏洩するのを抑制できる。

【００２９】本発明の請求項８に記載の発明は、第一の膨張機構あるいは第二の膨張機構を第一の吸入管あるいは第二の吸入管と熱交換する複数のキャピラリで形成し、冷蔵サイクル用液管あるいは冷凍サイクル用液管を複数の前記キャピラリで代用し、複数のキャピラリの流路を切り替えることで抵抗を変化させることを特徴とする請求項４記載の冷蔵庫であるので、冷蔵庫の標準的な冷却条件における安定運転時において冷蔵領域冷却時の高蒸発温度化と冷凍領域冷却用膨張機構の入口冷媒乾き度の低下により省エネルギー化を維持しながら、電源投入時に効率良く急冷ができることに加えて、液管を小ボリュームであるキャピラリで代用することで冷媒封入量が低減できる。

【００３０】本発明の請求項９に記載の発明は、冷蔵領

域と冷凍領域を備えた冷蔵庫であって、圧縮機と、凝縮器と、流路切替弁と、第一のキャピラリと、第二のキャピラリと、第三の蒸発器と、前記第一のキャピラリ及び第二のキャピラリと熱交換する第三の吸入管と、前記第三の蒸発器と前記冷蔵領域内の空気を熱交換する第一の風路と、前記第三の蒸発器と冷凍領域内の空気を熱交換する第二の風路とを備え、前記圧縮機と前記凝縮器と前記流路切替弁と前記第一のキャピラリと前記第三の蒸発器と前記第三の吸入管とを閉ループで構成すると共に、前記第一のキャピラリと並列になるように前記第二のキャピラリとを接続し、前記流路切替弁によりキャピラリへの冷媒の流れを切り替えることにより、前記第一の風路と前記第二の風路を開く時は前記第一のキャピラリを使用し、第二の風路のみを開く時は前記第二のキャピラリを使用して冷媒の流量を可変することを特徴とする冷蔵庫であるので、電源投入時等の過負荷時において抵抗の小さい第一のキャピラリで冷蔵領域と冷凍領域を同時に冷却する冷却運転と冷凍領域のみを冷却する冷却運転を交互に行うことで効率良く急冷が行えると共に、冷蔵領域と冷凍領域を同時に冷却することで冷蔵庫の標準的な冷却条件における安定運転時において冷蔵領域冷却時間を長くして冷蔵領域内の温度変動を抑制できる。

【００３１】本発明の請求項１０に記載の発明は、圧縮機は回転数可変型であり、第一の膨張機構と第二の膨張機構は絞り量が変化可能であり、外気温度を検知する外気温センサを有し、前記第一の膨張機構と第二の膨張機構の絞り量は前記外気温センサが検知した外気温から算出した負荷量に相当する必要冷媒流量が流通するように制御され、前記圧縮機の回転数は前記必要冷媒流量から所定蒸発温度になるように制御することを特徴とする請求項４から９のいずれか一項記載の冷蔵庫であるので、電源投入時等の過負荷時において効率良く急冷が行えると共に、常に冷媒流量に対応した熱交換能力を得ることができる蒸発温度となり冷凍サイクルの最大能力を使用して効率良く冷却を行う。

【００３２】本発明の請求項１１に記載の発明は、凝縮器と流路切替弁の間に受液器を設けた請求項４から１０のいずれか一項記載の冷蔵庫であるので、低冷媒循環量の冷凍領域冷却から高冷媒循環量の冷蔵領域冷却に切り替わる時の一時的な冷媒循環量不足を解消して早期に冷蔵室冷却の高効率サイクルに移行できる。

【００３３】

【発明の実施の形態】本発明による実施の形態１について、図面を参照しながら説明する。なお、従来例と同一構成については、同一符号を付して詳細な説明を省略する。

【００３４】（実施の形態１）図１は本発明の実施の形態１による冷却サイクル及び冷蔵庫の概略図である。

【００３５】図１において冷蔵庫の標準的な冷却条件における安定運転時の冷凍室２からなる冷凍領域の吸熱負荷量は、冷蔵室１からなる冷蔵領域の吸熱負荷量は略同一である。

【００３６】図１において、１４は逆止弁、１５は冷蔵室冷却時に冷媒が流通する冷蔵サイクル用液管、１６は第一の膨張機構、１７は第一の蒸発器５と圧縮機３を接続する第一の吸入管、１８は冷蔵サイクル用液管１５と第一の吸入管１７が熱交換する第一の熱交換部、１９は冷凍室冷却時に冷媒が流通する冷凍サイクル用液管、２０は流量可変型である第二の膨張機構、２１は第二の蒸発器６と圧縮機３とを接続する第二の吸入管、２２は冷凍サイクル用液管１９と第二の吸入管２１が熱交換する第二の熱交換部である。

【００３７】以上のように構成された冷蔵庫について、以下にその動作を説明する。

【００３８】冷蔵室１の冷却時は、図示していない冷蔵室１の庫内温度センサにより庫内温度を検知して所定温度以上になると、圧縮機３の運転により冷媒が圧縮され、圧縮された高温高圧の冷媒は凝縮器４で冷却されることで凝縮して流路切替弁９に流れる。その冷媒は出口側を冷蔵サイクル用液管１５に流通するように制御された流路切替弁９から冷蔵サイクル用液管１５に流通し、冷媒は冷蔵サイクル用液管１５を通る時に第一の熱交換部１８で第一の吸入管１７と熱交換して冷却されて過冷却状態となって第一の膨張機構１６に送られる。そして、冷媒は第一の膨張機構１６によりに減圧され蒸発することで冷凍室２の冷却時の蒸発温度よりは高い蒸発温度の低温となって第一の蒸発器５を流れる。このとき、冷蔵室１内の空気は第一のファン１０の作動により低温となった第一の蒸発器５と熱交換することで冷却されて循環して冷蔵室１内の冷却を行う。そして、第一の蒸発器５内の冷媒は乾き度を増しながら流通し、第一の蒸発器５の出口では飽和ガスとなって第一の吸入管１７の入口に至る。この冷媒は第一の吸入管１７を通る時に第一の熱交換部１８にて高温の冷蔵サイクル用液管１５と熱交換することで加熱されて適度なガスとなり圧縮機３に吸入される。このとき、冷凍室２の冷却用の第二の蒸発器６は冷凍室２の室温程度であり、第一の蒸発器５の蒸発圧力より低いが、逆止弁１４により冷媒の逆流は防止されている。

【００３９】冷凍室２の冷却時は、図示していない冷凍室２の庫内温度センサにより庫内温度を検知して所定温度以上になると、圧縮機３の運転により冷媒が圧縮され、圧縮された高温高圧の冷媒は凝縮器４で冷却されることで凝縮して流路切替弁９に流れる。その冷媒は出口側を冷凍サイクル用液管１９に流通するように制御された流路切替弁９から冷凍サイクル用液管１９に流通し、冷媒は冷凍サイクル用液管１９を通る時に第二の熱交換部２２で第二の吸入管２１と熱交換して冷却されて過冷却状態となって第二の膨張機構２０に送られる。そして、冷媒は第二の膨張機構２０によりに減圧され蒸発す

ることで低蒸発温度となって第二の蒸発器６を流れる。このとき、冷凍室２内の空気は第二のファン１１の作動により低温となった第二の蒸発器６と熱交換することで冷却されて循環して冷蔵室１内の冷却を行う。そして、第一の蒸発器５内の冷媒は乾き度を増しながら流通し、第二の蒸発器６の出口では飽和ガスとなって第二の吸入管２１の入口に至る。この冷媒は第二の吸入管２１を通る時に第二の熱交換部２２にて高温の冷凍サイクル用液管１９と熱交換することで加熱されて適度なガスとなり圧縮機３に吸入される。

【００４０】ここで、冷蔵庫の標準的な冷却条件における安定運転時においては、圧縮機３を最低回転数で運転すると共に、第二の膨張機構２０は第一の膨張機構１６に対して抵抗が２倍となるように調整して、冷凍室２冷却時の蒸発温度を－３０℃、冷蔵室１の冷却時の蒸発温度を－１５℃に制御している。このとき、冷蔵室１冷却時の冷媒循環量は冷凍室２冷却時の約２倍となるので、冷蔵室１の冷却運転時間を冷凍室２の冷却運転時間の約１／２倍とすることで、冷蔵領域と冷凍領域の吸熱負荷量比に対応した冷凍能力に調整することができる。

【００４１】このとき、適当な最低回転数での冷凍能力を有する圧縮機３を選定すれば、冷蔵室１の冷却運転と冷凍室２の冷却運転を切り替えながら、圧縮機３をほぼ連続運転することができる。この場合、総運転率１００％に対して、冷蔵室１の運転率は約３３％、冷凍室２の運転率は約６７％となる。また、冷蔵室１と冷凍室２の冷却運転の切り替えを頻繁に行うと冷媒流路を切り替えるロスが大きくなるため、冷蔵室１の冷却＋冷凍室２の冷却＋冷却停止の１サイクルを５０～１００分とすることが望ましい。このとき、冷蔵室１の１サイクル中の運転時間は約１７～３３分、冷凍室２の１サイクル中の運転時間は約３３～６７分となる。この結果、冷蔵室１と冷凍室２の冷却運転はともに問題なく効率の良い運転ができる。

【００４２】ここで、冷蔵室１あるいは冷凍室２の運転率が１５％以下になると、温度変動の抑制が困難となるとともに、冷蔵室１あるいは冷凍室２の運転時間が１０分以下になると、冷媒流路切り替えあるいは圧縮機起動直後に第一の蒸発器５内の冷媒が不足した状態で圧縮機を運転する冷却ロスの割合が大きくなり効率的な運転が困難になる。冷凍領域と冷蔵領域を切り替えて冷却するシステムの場合、単位時間あたりの冷凍能力が高い冷蔵領域の冷却運転の運転時間が短くなる傾向があり、冷凍領域と冷蔵領域の吸熱負荷量の設計が重要となる。

【００４３】理想的には、冷蔵庫の標準的な冷却条件における安定運転時の冷凍室２からなる冷凍領域の吸熱負荷量を冷蔵室１からなる冷蔵領域の吸熱負荷量の約１／２倍とすることが望ましい。この場合、冷蔵室１と冷凍室２の運転率はともに約５０％、運転時間も２５～５０分に調整できるため、運転率低下や運転時間低下の問題は生じない。一方、冷蔵庫の標準的な冷却条件における安定運転時の冷凍室２からなる冷凍領域の吸熱負荷量が冷蔵室１からなる冷蔵領域の吸熱負荷量の約３倍程度となると、冷蔵室１の運転率は約１４％、運転時間も７～１４分となり、温度変動の抑制や冷媒流路切り替えロスの抑制が困難となる。従って、冷蔵庫の標準的な冷却条件における安定運転時の冷凍領域の吸熱負荷量は、冷蔵領域の吸熱負荷量の１／２倍から２倍程度が望ましい。

【００４４】以上のように、冷蔵庫の標準的な冷却条件における安定運転時の冷凍室２からなる冷凍領域の吸熱負荷量を冷蔵室１からなる冷蔵領域の吸熱負荷量と略同一とすることにより、冷蔵室１の運転時間を確保することができ、冷蔵室１の温度変動や切り替え時の冷却ロスの割合を抑制することができる。

【００４５】（実施の形態２）本発明による実施の形態２について、図面を参照しながら説明する。なお、実施の形態１と同一構成及び作用については、同一符号を付して詳細な説明を省略する。

【００４６】図２は本発明の実施の形態２による冷却サイクル及び冷蔵庫の概略図である。

【００４７】図２において、断熱材１３は通常使用される熱伝導率０．０１５Ｗ／ｍＫのウレタン断熱材であり、４０は熱伝導率が０．００３Ｗ／ｍＫである高断熱性能の真空断熱材であり、外箱表面積の約５０％を真空断熱材４０で被覆している。そして、冷蔵庫の標準的な冷却条件における安定運転時の冷凍室２からなる冷凍領域の吸熱負荷量は１３Ｗ、冷蔵室１からなる冷蔵領域の吸熱負荷量は２７Ｗである。また、真空断熱材４０は、例えば、特開昭６０－１４６９９４公報に開示されているような内部に減圧脱気し外側を通気性にない袋で包んだ断熱材パックからなる。

【００４８】以上のように構成された冷蔵庫について、以下にその動作を説明する。

【００４９】冷蔵庫の標準的な冷却条件における安定運転時において、冷蔵室１は外気から断熱材１３を通して熱が侵入する。そして、この外気から侵入してくる熱量は圧縮機３を運転して第一の蒸発器５で冷媒を蒸発させることで取り去り、冷蔵室１内を５℃に保つ。また、冷凍室２は外気から断熱材１３と真空断熱材４０を通して室内に熱が侵入する。そして、この外気から侵入してくる熱量は圧縮機３を運転して第二の蒸発器６で冷媒を蒸発させることで取り去り、冷凍室２内を－２０℃に保つ。このとき、真空断熱材４０の断熱効果により冷凍室２からなる冷凍領域の吸熱負荷量を冷蔵室１からなる冷蔵領域の吸熱負荷量の約１／２倍に抑制することができ、冷蔵室１と冷凍室２の運転率は約５０％に設計できる。

【００５０】ここで、冷凍室２の外周の断熱に通常の断熱材１３のみを使用した場合と断熱材１３と真空断熱材４０を積層した場合の壁厚を（表１）に示す。

【００５１】 ＊ ＊【表１】

	合計厚さ	ウレタン断熱材		真空断熱材		熱通過率
		厚さ	熱伝導率	厚さ	熱伝導率	
	mm	mm	W/mK	mm	W/mK	W/m2K
実施の形態2	64	32	0.015	32	0.003	0.078
従来例	193	193	0.015	0	0.003	0.078

【００５２】（表１）において、負荷量は２５℃の外気あるいは冷蔵室１から断熱材１３や真空断熱材４０を通して冷凍室２内に侵入してくる吸熱負荷量である。（表１）に示したように、冷凍室２は真空断熱材４０を使用することで壁厚が薄い状態でも侵入熱量を極端に小さくでき、侵入熱量の多い冷蔵室１と同じ壁厚で設計が可能である。

【００５３】以上のように、冷凍室２の外周の断熱に真空断熱材を使用することにより、薄い壁厚を維持しながら、冷蔵庫の標準的な冷却条件における安定運転時の冷凍室２からなる冷凍領域の吸熱負荷量を冷蔵室１からなる冷蔵領域の吸熱負荷量の１／２倍とすることにより、冷蔵室１の運転時間を確保することができ、冷蔵室１の温度変動や切り替え時の冷却ロスの割合を抑制することができる。

【００５４】なお、本実施の形態では冷蔵庫の外箱表面積の約５０％を真空断熱材４０で被覆したが、被覆率は５０～８０％が望ましい。被覆率が５０％より小さい場合、冷凍領域全体を被覆できず吸熱負荷の低減が困難であるとともに、被覆率が８０％より大きい場合、外箱角部等において真空断熱材４０の突合せにより通常の断熱材１３が薄肉となり構造強度の低下が問題となる。また、真空断熱材４０は平面部の方が設置しやすいため、圧縮機３等が設置されている機械室の上部と冷凍室２の境界部の断熱材１３を平面形状にする方が望ましい。

【００５５】（実施の形態３）本発明による実施の形態３について、図面を参照しながら説明する。なお、従来と同一構成及び動作については、同一符号を付して詳細な説明を省略する。

【００５６】図３は本発明の実施の形態３による冷却サイクル及び冷蔵庫の概略図である。

【００５７】図３において、１４は逆止弁、１５は冷蔵室冷却時に冷媒が流通する冷蔵サイクル用液管、１６は第一の膨張機構、１７は第一の蒸発器５と圧縮機３を接続する第一の吸入管、１８は冷蔵サイクル用液管１５と第一の吸入管１７が熱交換する第一の熱交換部、１９は冷凍室冷却時に冷媒が流通する冷凍サイクル用液管、２０は流量可変型である第二の膨張機構、２１は第二の蒸発器６と圧縮機３とを接続する第二の吸入管、２２は冷凍サイクル用液管１９と第二の吸入管２１が熱交換する第二の熱交換部である。

【００５８】以上のように構成された冷蔵庫について、以下にその動作を説明する。

【００５９】冷蔵室１の冷却時は、図示していない冷蔵室１の庫内温度センサにより庫内温度を検知して所定温度以上になると、圧縮機３の運転により冷媒が圧縮され、圧縮された高温高圧の冷媒は凝縮器４で冷却されることで凝縮して流路切替弁９に流れる。その冷媒は出口側を冷蔵サイクル用液管１５に流通するように制御された流路切替弁９から冷蔵サイクル用液管１５に流通し、冷媒は冷蔵サイクル用液管１５を通る時に第一の熱交換部１８で第一の吸入管１７と熱交換して冷却されて過冷却状態となって第一の膨張機構１６に送られる。そして、冷媒は第一の膨張機構１６によりに減圧され蒸発することで冷凍室２の冷却時の蒸発温度よりは高い蒸発温度の低温となって第一の蒸発器５を流れる。このとき、冷蔵室１内の空気は第一のファン１０の作動により低温となった第一の蒸発器５と熱交換することで冷却されて循環して冷蔵室１内の冷却を行う。そして、第一の蒸発器５内の冷媒は乾き度を増しながら流通し、第一の蒸発器５の出口では飽和ガスとなって第一の吸入管１７の入口に至る。この冷媒は第一の吸入管１７を通る時に第一の熱交換部１８にて高温の冷蔵サイクル用液管１５と熱交換することで加熱されて適度なガスとなり圧縮機３に吸入される。このとき、冷凍室２の冷却用の第二の蒸発器６は冷凍室２の室温程度であり、第一の蒸発器５の蒸発圧力より低いが、逆止弁１４により冷媒の逆流は防止されている。

【００６０】冷凍室２の冷却時は、図示していない冷凍室２の庫内温度センサにより庫内温度を検知して所定温度以上になると、圧縮機３の運転により冷媒が圧縮され、圧縮された高温高圧の冷媒は凝縮器４で冷却されることで凝縮して流路切替弁９に流れる。その冷媒は出口側を冷凍サイクル用液管１９に流通するように制御された流路切替弁９から冷凍サイクル用液管１９に流通し、冷媒は冷凍サイクル用液管１９を通る時に第二の熱交換部２２で第二の吸入管２１と熱交換して冷却されて過冷却状態となって第二の膨張機構２０に送られる。そして、冷媒は第二の膨張機構２０によりに減圧され蒸発することで低蒸発温度となって第二の蒸発器６を流れる。このとき、冷凍室２内の空気は第二のファン１１の作動により低温となった第二の蒸発器６と熱交換することで冷却されて循環して冷蔵室１内の冷却を行う。そして、第一の蒸発器５内の冷媒は乾き度を増しながら流通し、第二の蒸発器６の出口では飽和ガスとなって第二の吸入管２１の入口に至る。この冷媒は第二の吸入管２１を通る時に第二の熱交換部２２にて高温の冷凍サイクル用液管１９と熱交換することで加熱されて適度なガスとなり

圧縮機３に吸入される。

【００６１】ここで、通常運転時においては、圧縮機３を最低回転数で運転するとともに、第二の膨張機構２０は第一の膨張機構１６に対して抵抗が約２倍となるように調整して、冷凍室２の冷却時の蒸発温度を－３０℃、冷蔵室１の冷却時の蒸発温度を－１５℃に制御している。

【００６２】そして、電源投入時においては圧縮機３を最高回転数で運転し、第二の膨張機構２０の抵抗を第一の膨張機構１６の抵抗と同等程度になるように制御する。この結果、冷凍室２冷却時の冷媒流量を冷蔵室１冷却時の冷媒流量と同程度まで増加させて、急冷することができる。このとき、第一の蒸発器５と第二の蒸発器６の蒸発温度は増加した冷媒流量に対応するため－２０～－３０℃に設定する方が望ましい。これ以上に抵抗を小さくすると冷媒流量が増加すると同時に蒸発温度が上昇し、蒸発器での熱交換温度差が小さくなるため蒸発器内の冷媒を蒸発しきれず無駄となる。また、冷凍室２冷却および冷蔵室１冷却ともに、冷却システムの最大能力を使うことから電源投入後からのプルダウン時間を最短にすることができる。

【００６３】なお、第二の蒸発器６を除霜した後の運転等の過負荷時においても、電源投入時と同様に第二の膨張機構２０の抵抗を制御しても冷凍室２を急冷する効果が得られる。

【００６４】以上のように、電源投入時等の過負荷時において、第二の膨張機構２０の抵抗を第一の膨張機構１６の抵抗と同等程度になるように制御することで効率よく急冷ができる。

【００６５】（実施の形態４）本発明による実施の形態４について、図面を参照しながら説明する。なお、実施の形態３と同一構成及び動作については、同一符号を付して詳細な説明を省略する。

【００６６】本実施の形態における構成の特徴は、冷蔵サイクル用液管１５を内径１．２mm、長さ２．４m、冷凍サイクル用液管１９を内径０．８mm、長さ２．４mの内面が滑らかな銅管で形成した点である。冷蔵サイクル用液管１５および冷凍サイクル用液管１９は、凝縮器４で液化された冷媒をそれぞれ第一の膨張機構１６および第二の膨張機構２０へ供給するとともに、それぞれ第一の熱交換部１８および第二の熱交換部２２において第一の吸入管１７および第二の吸入管２１と熱交換するものである。

【００６７】一般に、冷蔵サイクル用液管１５および冷凍サイクル用液管１９は内径３～４mmの銅製の細径管が用いられるが、Ｒ６００ａやＲ２９０等の可燃性冷媒を使用する場合、内部に保持される液冷媒量が１０～２０ｇと大きくなることから、使用冷媒量を削減する観点からより細径化が望まれている。本実施の形態では、細径化の限界を明らかにするとともに、切替システムを搭載した冷蔵庫の通常運転時と電源投入時を考慮した最適な内径量を提案するものである。

【００６８】図４は本実施の形態による冷却サイクルのＰ－ｈ線図である。図４で示したものは、最も循環量が大きい高外気温の電源投入時の冷蔵室冷却サイクルである。

【００６９】図４において、Ａは冷蔵サイクル用液管１５の入口における冷媒の状態、Ｂは第一の膨張機構１６の入口における冷媒の状態、Ｃは第一の膨張機構１６の出口であり第一の蒸発器５の入口における冷媒の状態、Ｄは第一の蒸発器５の出口であり第一の吸入管１７の入口における冷媒の状態、Ｅは第一の吸入管１７の出口であり圧縮機３の吸入部における冷媒の状態であり、冷蔵サイクル用液管１５と第一の吸入管１７は第一の熱交換部１８にて１００％熱交換される。これにより、圧縮機３の吸入部のエンタルピーと第一の吸入管１７の入口のエンタルピーとの差が冷蔵サイクル用液管１５の入口のエンタルピーと第一の膨張機構１６の入口のエンタルピーとの差と等しくなる。つまり、ＥとＤのエンタルピー差がＡとＢのエンタルピー差と等しい。

【００７０】本実施の形態の冷蔵サイクル用液管１５は、内径１．２mm、長さ２．４mの細径管を使用しているため、管内を流通する冷媒Ｒ６００ａの液量を２～３ｇと極少量に抑えることができる。しかし、細径管のため管内抵抗による圧損が生じ、Ｂ点で示したように第一の膨張機構１６の入口における冷媒の圧力が、Ａ点で示した冷蔵サイクル用液管１５の入口の圧力より低下する。この内径ではＢ点は過冷却域にあり、膨張機構１６の動作に不具合は生じないが、内径を０．８mmまで絞ると管内抵抗による圧損が生じ、最も大きい循環量を示す高外気温の電源投入時のの条件では図４のＢ１点で示したように過冷却０℃ギリギリの状態となる。さらに内径を小さくすると、圧損が増加し図４のＢ２点で示したように２相域に移行して、膨張機構１６の動作が不安定になるとともに、見かけ上の膨張機構１６の抵抗値が増加して冷媒流量が著しく低下する問題が発生する。

【００７１】また、本実施の形態の冷凍サイクル用液管１９も同様にして、内径０．８mm、長さ２．４mの細径管を使用しているため、管内を流通する冷媒Ｒ６００ａの液量を１ｇ以下と極少量に抑えることができるとともに、最も大きい循環量を示す高外気温の電源投入時の条件においても２相域に移行することなく冷媒を流通させることができる。なお、第二の膨張機構２０の抵抗を大きくして制御する通常運転時においては、冷凍サイクル用液管１９の出口は冷蔵サイクルと同様に過冷却域の状態となる。

【００７２】以上のように、電源投入時等の過負荷時において、第二の膨張機構の抵抗を第一の膨張機構の抵抗と同等程度になるように制御することで効率よく急冷ができるとともに、冷蔵サイクル用液管および冷凍サイク

ル用液管の内径を０．８ｍｍ以上とすることで、管内を流通する冷媒の液量を極少量に抑えながら、第一の膨張機構および第二の膨張機構の流量制御を安定して行うことができる。

【００７３】（実施の形態５）本発明による実施の形態５について、図面を参照しながら説明する。なお、実施の形態４と同一構成及び動作については、同一符号を付して詳細な説明を省略する。

【００７４】図５は本発明の実施の形態４による冷却サイクル及び冷蔵庫の概略図、図６は要部の斜視断面図である。図５及び図６において、２３と２４は第一の液管と第二の液管である。

【００７５】本実施の形態における構成の特徴は、第一の液管２３と第二の液管２４の流路を並行に形成し、それぞれを内径０．５７ｍｍ、長さ１．２ｍの内面が滑らかな銅管で形成した点である。第一の液管２３と第二の液管２４は、凝縮器４で液化された冷媒をそれぞれ第二の膨張機構２０へ供給するとともに、第二の熱交換部２２において第二の吸入管２１と熱交換するものであるこれによって、冷媒流量を確保したまま、第一の液管２３および第二の液管２４と第二の吸入管２１との熱交換に必要な長さを短くするとともに、流路抵抗を低減することができ、より細い管径で冷媒の過冷却が確保できる。この結果、Ｒ６００ａやＲ２９０等の可燃性冷媒を使用する場合、内部に保持される液冷媒量を削減することができる。

【００７６】なお、本実施の形態では第一の液管２３と第二の液管２４を内径０．５７ｍｍ、長さ１．２ｍとしたが、内径０．５ｍｍ以上の２本以上の液管であれば同様の効果が期待できる。また、冷蔵室１の冷却用サイクルにおいても同様の効果が得られる。

【００７７】以上のように、電源投入時等の過負荷時において、第二の膨張機構の抵抗を第一の膨張機構の抵抗と同等程度になるように制御することで効率よく急冷ができるとともに、冷蔵サイクル用液管あるいは冷凍サイクル用液管を内径０．５ｍｍ以上の複数の液管で形成することで、熱交換に必要な長さを短くするとともに、管内を流通する冷媒の液量を極少量に抑えながら、第一の膨張機構および第二の膨張機構の流量制御を安定して行うことができる。

【００７８】（実施の形態６）本発明による実施の形態６について、図面を参照しながら説明する。なお、実施の形態３と同一構成及び動作については、同一符号を付して詳細な説明を省略する。

【００７９】図７は本発明の実施の形態６による冷却サイクル及び冷蔵庫の概略図である。

【００８０】図７に示すように、２５は第一の膨張弁、２６は第一の膨張弁２５を冷蔵室１の空気と隔離するための第一の隔離壁、２７は第二の膨張弁、２８は第二の膨張弁２７を冷凍室２の空気と隔離するための第二の隔離壁である。図示していないが、第一の隔離壁２６と第二の隔離壁２８は難燃性樹脂から構成され、膨張弁が破損した場合に交換が可能なように一部が開閉できる構造となっている。さらに、設置場所は室内から隠れた第一の蒸発器５と第二の蒸発器６の近傍であると共に、各蒸発器と熱交換する室内空気の抵抗とならず、図示していない蒸発器の除霜ヒータにより外郭を除霜可能な位置である。

【００８１】本実施の形態における構成の特徴は、第一の膨張弁２５および第二の膨張弁２７をそれぞれ第一の隔離壁２６と第二の隔離壁２８で囲うことにより、冷蔵庫室内の空気との接触を抑制することにある。これにより、外部からの受熱を抑制して第一の膨張弁２５および第二の膨張弁２７の流量調整を安定させるとともに、接合部等から漏洩が生じた場合、食品への悪影響を低減することができる。また、特にＲ６００ａやＲ３９０等の可燃性冷媒を用いた場合、冷蔵庫室内への漏洩を抑制し発火の危険性を低減できるという効果もある。

【００８２】また、本実施の形態では膨張弁は冷蔵庫の室内に設置しているが、隔離壁の断熱性能が十分であれば膨張弁への受熱を回避できるので室外に設置しても良い。

【００８３】以上のように、電源投入時等の過負荷時において、第二の膨張機構の抵抗を第一の膨張機構の抵抗と同等程度になるように制御することで効率よく急冷ができるとともに、膨張弁を隔離壁で囲うことで、漏洩時の食品等への悪影響が抑制できる。

【００８４】（実施の形態７）本発明による実施の形態７について、図面を参照しながら説明する。なお、実施の形態１と同一構成及び動作については、同一符号を付して詳細な説明を省略する。

【００８５】図８は本発明の実施の形態７による冷却サイクル及び冷蔵庫の概略図である。

【００８６】図８に示すように、２９は第三のキャピラリであり内直径が０．７７ｍｍで長さが２３１０ｍｍのキャピラリであり、第二の熱交換部２２で第二の吸入管２１と熱交換している。第一のキャピラリ７は内直径が０．７７ｍｍで長さが２３１０ｍｍ、第二のキャピラリ８は内直径が０．５６ｍｍで長さが２３１０ｍｍのキャピラリである。また、３０は冷媒流路を第一のキャピラリ７または第二のキャピラリ８または第三のキャピラリ２９に切り替える多方向切替弁であり、圧縮機３は回転数が２８ｒｐｓから７５ｒｐｓの可変型である。

【００８７】以上のように構成された冷蔵庫について、以下にその動作を説明する。

【００８８】通常冷却時と電源投入時等の過負荷時の各部の状態を従来例と本実施の形態を比較して（表２）に示す。

【００８９】

【表２】

圧縮機気筒容積:5.7mL 圧縮機吸入温度:30℃			キャピラリ仕様 (内径[mm]×長さ[mm])	蒸発温度	凝縮温度	圧縮機の回転数	体積効率	循環量
				℃	℃	rps	%	kg/h
実施の形態3	通常負荷	冷蔵	φ0.77×2310	−15	30	28	80	3.1
		冷凍	φ0.56×2310	−30	30	28	70	1.4
	電源投入	冷蔵	φ0.77×2310	−27	45	75	65	4.0
		冷凍	φ0.77×2310	−27	45	75	65	4.0
従来例	通常負荷	冷蔵	φ0.77×2310	−15	30	28	80	3.1
		冷凍	φ0.56×2310	−30	30	28	70	1.4
	電源投入	冷蔵	φ0.77×2310	−27	45	75	65	4.0
		冷凍	φ0.56×2310	−38	45	75	50	1.8

【００９０】（表２）の通り、通常の低負荷時には圧縮機３の回転数を最低の２８ｒｐｓに運転して、冷凍室２の冷却時は第一のキャピラリ７より抵抗の大きい第二のキャピラリ８に冷媒を流通させることで吸入ガス密度が小さくなる第二の蒸発器６の蒸発温度を－３０℃、吸入ガス密度が大きい冷第一の蒸発器５の蒸発温度を－１５℃とすることで従来と同等の省エネルギーサイクルを維持する。

【００９１】そして、負荷が急増する電源投入時のような過負荷時は圧縮機３を最高回転数の７５ｒｐｓで運転し、低冷媒循環量である冷凍室２冷却時に冷蔵室１冷却時と同抵抗の第三のキャピラリ２９に冷媒を流通させることで冷媒循環量を増加させて高冷凍能力を得ることで従来より速く冷却ができる。このとき、第一の蒸発器５と第二の蒸発器６の蒸発温度は－２７℃とすることで増加した冷媒流量に対応する熱交換能力を得るので、冷却システムの最大冷凍能力を使用することができプルダウン時間を最短にできる。

【００９２】さらに、液管を小ボリュームであるキャピラリで代用しているので冷媒封入量が低減でき経済的であると共に、膨張弁等に比べて安価に冷媒漏洩時の食品への悪影響の防止や可燃性冷媒を用いて漏洩した場合の発火の危険性を低減できる。

【００９３】なお、電源投入時と同様に、外気温上昇により負荷が増加する高負荷時においても、外気温センサ等を用いて高外気温を検知した場合に第三のキャピラリ２９に切り替えることで同様の効果は得られる。

【００９４】また、本実施の形態では複数のキャピラリは２本であるが、それ以上であれば更に広範囲で流量制御ができるので、同様以上の効果は得られる。また、冷蔵室１の冷却サイクル側に設置しても良い。

【００９５】また、本発明では多方向切替弁３０にて複数のキャピラリから１本のキャピラリへ流路を切り替えているが、最大抵抗のキャピラリ以外のキャピラリ前後に開閉弁を設置して必要に応じて開閉する構成でも同様の効果は実現できる。

【００９６】また、本実施の形態では抵抗差の違う第二のキャピラリ８と第三のキャピラリ２９を冷凍室２の冷却時に必要に応じてどちから一方に冷媒を流通させているが、同一抵抗のキャピラリを２本用いて必要に応じて両方同時か片方のみに冷媒を流通させることで流量制御を行っても同様の効果は得られ、その他のキャピラリ複数を用いて流通切替を行うことで必要に応じて所定の流量を流せる様に流量可変制御ができるのならば同様の効果が得られる。

【００９７】（実施の形態８）本発明による実施の形態８について、図面を参照しながら説明する。なお、従来と同一構成及び動作については、同一符号を付して詳細な説明を省略する。

【００９８】図９は本発明の実施の形態８による冷却サイクル及び風路構成の概略図である。

【００９９】図９において、３１は第三の蒸発器、３２は第三の吸入管、３３は第一のキャピラリ７及び第二のキャピラリ８が第三の吸入管３２と熱交換する第三の熱交換部、３４は第三の蒸発器３１と熱交換後の空気を冷蔵室１または冷凍室２に循環させるためのファン、３５は冷凍室２と冷蔵室１を連通し冷凍室２の空気を冷蔵室１に吐出する冷蔵室吐出ダクト、３６は第三の蒸発器３１と熱交換後の空気が冷凍室２に導びかれる冷凍室吐出ダクト、３７は冷蔵室１内の空気を第三の蒸発器３１に導く冷蔵室吸入ダクト、３８は冷凍室２内の空気を第三の蒸発器３１に導く冷凍室吸入ダクト、３９は第三の蒸発器３１と熱交換後の低温空気を冷蔵室吐出ダクト３５または冷凍室吐出ダクト３６に風路を切り替えるダンパであり、矢印は通風方向を示している。

【０１００】また、図示していないが、冷凍室吐出ダクト３６と冷凍室２とが連通する付近には、第三の蒸発器３１と熱交換後の吐出空気温度を検知する吐出空気温度センサを設けている。

【０１０１】以上のように構成された冷蔵庫について、以下にその動作を説明する。

【０１０２】本実施の形態における構成の特徴は、通常運転時においては第一のキャピラリ７を用いて冷凍室２と冷蔵室１を同時に冷却する同時冷却モードと、第二のキャピラリ８を用いて冷凍室２のみを冷却する冷凍室冷却モードを切り替えて冷却し、電源投入時においては第一のキャピラリ７のみを用いて同時冷却モードと冷凍室冷却モードの運転を行うことにある。

【０１０３】通常運転時においては、まず、流路切替弁９により抵抗の小さい第一のキャピラリ７に冷媒が流通するように制御され、ダンパ３９が開きファン３４の作動により第三の蒸発器３１と熱交換した空気は冷凍室吐

出ダクト３６を通って冷凍室２内に吐出し、主として冷蔵室吐出ダクト３５を通って冷蔵室に吐出され冷蔵室吸入ダクト３７を通って第三の蒸発器３１に通風するように循環する。これにより、冷凍室２と冷蔵室１を同時に冷却する同時冷却モードとなる。このとき、蒸発温度が－２２℃程度になるように圧縮機３の回転数を調整する。

【０１０４】次に、流路切替弁９により抵抗の大きい第二のキャピラリ８に冷媒が流通するように制御され、ダンパ３９が閉じてファン３４の作動により第三の蒸発器３１と熱交換した空気は冷凍室吐出ダクト３６から冷凍室２に吐出し、冷凍室吸入ダクト３８を通って第三の蒸発器３１と熱交換するように循環する。これにより、冷凍室２のみを冷却する冷凍室冷却モードとなる。このとき、蒸発温度が－３０℃程度になるように圧縮機３の回転数を調整する。

【０１０５】以下、同時冷却モードと冷凍室冷却モードを交互に切り替えながら冷却する。このとき、冷蔵室１が所定の温度になれば同時冷却モードの運転を中止し、冷凍室２が所定の温度になれば冷凍室冷却モードの運転も中止する。

【０１０６】電源投入時においては、圧縮機３を最高回転数で運転するとともに、抵抗の小さい第一のキャピラリ７を用いて同時冷却モードと冷凍室冷却モードの冷却運転を交互に行う。このとき、蒸発温度が－２７℃程度になるように圧縮機３の回転数を調整する。そして、冷凍室２が所定の温度になれば冷却運転を中止するとともに、通常運転の制御に切り替える。

【０１０７】この結果、通常運転時の同時冷却モードにおいては、冷凍室冷却モードに比べて熱交換する空気温度が高いため理論効率の高い高蒸発温度での冷却が可能となり、総合的な冷却効率を向上することができる。また、冷蔵室１単独の冷却モードに比べると蒸発温度は低くなるが、冷却運転時間を長く設定できる利点がある。これは、冷蔵室１単独の冷却モードに比べると熱交換する空気温度が低く、また冷凍室２の空気温度以上の蒸発温度では冷凍室２を加温する可能性があることから、同時冷却モードの蒸発温度は－２０℃前後が限界となるためである。

【０１０８】さらに、電源投入時においては、冷蔵室１および冷凍室２ともに冷却システムの最大能力を使って冷却することから電源投入後からのプルダウン時間を最短にすることができる。

【０１０９】なお、第三の蒸発器３１を除霜した後の運転等の過負荷時においても、電源投入時と同様に第一のキャピラリ７を用いて冷凍室冷却モードを実行しても冷凍室２を急冷する効果が得られる。また、冷凍室冷却モードにおいて食品投入等の負荷の急増した場合、第三の蒸発器３１と熱交換後の吐出空気温度の上昇を検知して、第一のキャピラリ７に切り替えるとともに圧縮機３の回転数を増加させて蒸発温度を維持すれば、同様に冷凍室２を急冷する効果が得られる。

【０１１０】以上のように、電源投入時等の過負荷時において、抵抗の小さい第一のキャピラリを用いて同時冷却モードと冷凍室冷却モードの冷却運転を交互に行うことで効率よく急冷ができるとともに、同時冷却モードで冷蔵室を冷却することで、通常運転時の冷蔵室運転時間を長くして冷蔵室の温度変動を抑制することができる。

【０１１１】（実施の形態９）本発明による実施の形態９について、図面を参照しながら説明する。なお、実施の形態３と同一の構成および作用については、同一符号を付して詳細な説明を省略する。

【０１１２】本発明の実施の形態９による冷却サイクルおよび冷蔵庫は図１で示した実施の形態１と同一である。また、図１０は第一の蒸発器５と第二の蒸発器６の蒸発温度と蒸発能力の関係を示す図である。

【０１１３】図１０において、第一の蒸発器５の蒸発能力は所定の蒸発温度で、冷蔵室１の空気と熱交換して蒸発させることができる冷媒流量を示す。同様に、第二の蒸発器６の蒸発能力は所定の蒸発温度で、冷凍室２の空気と熱交換して蒸発させることができる冷媒流量を示す。第一の蒸発器５の蒸発能力と第二の蒸発器６の蒸発能力に大きな差があるのは、熱交換する空気温度の差によるところが大きい。従って、電源投入時のように熱交換する空気温度が高く大きな差がない場合は、第一の蒸発器５および第二の蒸発器６の蒸発能力はほぼ同等であり、図１０に示した第一の蒸発器５の蒸発能力よりも高い。

【０１１４】以下に本実施の形態の通常運転時における動作を説明する。

【０１１５】所定の外気温度における冷蔵庫の吸熱負荷に対応する冷却システムに必要な冷媒流量を設定し、予め外気温度と冷媒流量の関係を規定した制御テーブルを設定しておく。通常運転時には、外気温度センサー（図示せず）で検知した外気温度と前記制御テーブルから、目標とする冷媒流量を決定する。

【０１１６】ここで、所定の外気温度における冷蔵庫の吸熱負荷は、ドア開閉負荷や食品投入の負荷を含まない、冷蔵庫本体１２の断熱材１３を通じて流入する熱量を想定する方がより効率的な運転条件で制御できるので望ましい。また、予め規定しておく冷媒流量は、所定の吸熱負荷を運転率７０～８０％で冷却できる程度に設定すれば、比較的効率よくかつドア開閉負荷や食品投入等の変動要因を運転率の増加である程度対応できる。また、外気温度が１０℃以下で極めて吸熱負荷が小さく、冷却システムの低能力化が効率上好ましくない場合、運転率が低くなるように冷媒流量を設定してもよい。

【０１１７】次に、目標とする冷媒流量となるように、膨張機構１６と膨張機構２０の抵抗値および凝縮器４の能力を調整する。このとき、膨張機構１６あるいは膨張

機構２０の入口の冷媒状態が過冷却０℃近傍になることを想定して膨張機構１６と膨張機構２０の抵抗値を調整するとともに、大きな乾き度を持たないように凝縮器４の能力を調整することがサイクル効率上望ましい。
【０１１８】そして、目標とする冷媒流量において、第一の蒸発器５と第二の蒸発器６が最大能力を示す蒸発温度になるように圧縮機３の回転数を調整する。本実施の形態では図８のＡ点とＢ点で示した状態で第一の蒸発器５と第二の蒸発器６が動作する。ここで、冷蔵室１と冷凍室２では吸熱負荷の外気温依存性がことなること、また、第一の蒸発器５と第二の蒸発器６が最大能力大きく違うことから、冷蔵室１と冷凍室２それぞれ独立に圧縮機３の回転数を調整することが望ましい。
【０１１９】なお、ドア開閉負荷や食品投入等の吸熱負荷の変動要因が予測を超えて、運転率が１００％近くに達した場合、前記制御テーブルで規定された冷媒循環量の目標値を所定量増加させて、同様の制御を行えばよい。このとき、第一の蒸発器５あるいは第二の蒸発器６と熱交換された出口空気温度の変動から、急激な吸熱負荷の増加を検知して冷媒循環量の目標値を所定量増加させてもよい。
【０１２０】この結果、通常の使用条件である外気温度２５℃における吸熱負荷量に合わせて設定された第一の蒸発器５と第二の蒸発器６の蒸発温度、例えば－１５℃と－３０℃で固定的に運転制御された冷却システムに比べて、吸熱負荷に合わせて蒸発温度を変動させることにより特に吸熱負荷が小さい時に理論効率を最大限に高めることができ、冷蔵庫の通年の消費電力量を削減することができる。また、吸熱負荷の外気温依存性が異なる冷蔵室１と冷凍室２を独立に制御する切替システムを用いた高断熱性能の冷蔵庫においては特に消費電力量を削減する効果が大きい。
【０１２１】なお、本実施の形態においては、抵抗値が任意に可変できる膨張機構１６と膨張機構２０を用いて冷媒流量を制御したが、外気温度すなわち凝縮温度に対して適切に冷媒流量が変化するキャピラリ等の一定の抵抗を用いてもよいし、抵抗値の異なる複数のキャピラリを切り替えて冷媒流量を制御してもよい。
【０１２２】（実施の形態１０）本発明による実施の形態１０について、図面を参照しながら説明する。なお、実施の形態１及び実施の形態７と同一構成及び作用については、同一符号を付して詳細な説明を省略する。
【０１２３】図１１は本発明の実施の形態１０による冷却サイクル及び冷蔵庫の概略図であり、図１２は受液器の断面図及び冷蔵庫システムの概略図である。
【０１２４】図１１及び図１２にて、４１は凝縮器４と流路切替弁９の間に設けられた受液器である。
【０１２５】以上のように構成された冷蔵庫について、以下にその動作を説明する。
【０１２６】通常時における冷凍室２の冷却から冷蔵室１の冷却に切り替わる時は第二の膨張機構２０より絞り量の小さい第一の膨張機構１６のサイクルに移行する。このとき、受液器４１内に滞留していた液冷媒が冷蔵サイクル用液管１５を通って第一の膨張機構に流れて冷媒循環量が増加し、早期に所定の高冷媒循環量に安定する。
【０１２７】そして、電源投入時においては圧縮機３を最高回転数で運転し、第二の膨張機構２０の抵抗を第一の膨張機構１６の抵抗と同等程度になるように制御する。この結果、冷凍室２冷却時の冷媒流量を冷蔵室１冷却時の冷媒流量と同程度まで増加させると共に、冷媒流量に対応した熱交換能力を得る蒸発温度にすることで効率良く急冷を行う。
【０１２８】以上のように、電源投入時等の過負荷時に効率良く急冷を行うことができると共に、通常負荷における冷凍室２の冷却から冷蔵室１の冷却への切替時に、冷蔵室１の冷却時の所定の高冷媒循環量に必要な冷媒が受液器４１から流れ、早期に所定の高冷媒循環量に安定して圧縮機効率の良好である低圧縮比状態へ移行するので、圧縮機の消費電力が低減する。
【０１２９】
【発明の効果】以上説明したように本発明の請求項１に記載の発明は、断熱箱体内に冷蔵領域と冷凍領域を備えた冷蔵庫であって、前記冷蔵領域と前記冷凍領域にそれぞれ蒸発器を有し、少なくとも前記冷蔵領域の蒸発器に冷媒を流す冷媒回路と、前記冷凍領域の蒸発器に冷媒を流す冷媒回路とを設けてこれら冷媒回路を切り替えて冷却するものにおいて、冷蔵庫の標準的な冷却条件における安定運転時の前記冷凍領域の吸熱負荷量を前記冷蔵領域の吸熱負荷量と同等以下にしたので、比較的冷凍能力が低い冷凍領域の冷却運転時間を抑制することで、冷凍能力が大きい冷蔵領域の冷却運転時間を維持することができ、たとえば１５％以下の極端な低運転率になることが防止できるので、冷蔵領域の温度制御が容易になるとともに、圧縮機起動時の冷却ロスの割合を抑制し結果として効率的な運転が達成できる。
【０１３０】また、請求項２に記載の発明は、断熱箱体の断熱壁は発泡断熱材で形成され、冷凍領域の前記断熱壁には真空断熱材を配設したので、断熱壁を厚くせずに有効内容積を確保するとともに、冷凍能力が大きい冷蔵領域の冷却運転時間を維持することができる。
【０１３１】また、請求項３に記載の発明は、断熱箱体の断熱壁は発泡断熱材で形成され、前記断熱壁には外箱表面積の５０～８０％の範囲で真空断熱材を配設したので、断熱壁を厚くせずに有効内容積を確保するとともに、効果的に真空断熱材を配設することで高いコストパフォーマンスが得られる。
【０１３２】また、請求項４に記載の発明は、断熱箱体内に冷蔵領域と冷凍領域を備えた冷蔵庫であって、前記冷蔵領域に第一の蒸発器、前記冷凍領域に第二の蒸発器

を有し、圧縮機と、凝縮器と、流路切替弁と、冷蔵サイクル用液管と、前記第一の蒸発器と、前記冷蔵サイクル用液管と熱交換する第一の吸入管とを閉ループで構成するとともに、前記冷蔵サイクル用液管と前記第一の膨張機構と前記第一の蒸発器と前記第一の吸入管とに並列になるように冷凍サイクル用液管と、第二の膨張機構と、前記第二の蒸発器と、前記冷凍サイクル用液管と熱交換する第二の吸入管と、逆止弁とを接続し、前記流路切替弁により冷媒の流れを切り替えることで前記冷蔵領域と前記冷凍領域の冷却を互いに独立して行うものであり、電源投入時は前記第二の膨張機構の抵抗を冷蔵庫の標準的な冷却条件における安定運転時の抵抗より小さくすることを特徴とする冷蔵庫であるので、電源投入時等の過負荷運転時に冷凍領域冷却時は冷蔵領域冷却時と同等の高冷媒循環量として迅速に冷却状態に安定させることができる。

【０１３３】また、請求項５に記載の発明は、冷蔵サイクル用液管および冷凍サイクル用液管は内径が０．８ｍｍ以上であることを特徴とするので、電源投入時等の過負荷運転時に冷凍領域冷却時は冷蔵領域冷却時と同等の高冷媒循環量して急冷却を促進するとともに、冷蔵サイクル用液管あるいは冷凍サイクル用液管に滞留する冷媒の液量を少量に抑制して膨張機構の流量制御を安定して行うことができる。

【０１３４】また、請求項６に記載の発明は、冷蔵サイクル用液管あるいは冷凍サイクル用液管は並行した複数の液管で形成され、前記液管は内径が０．５ｍｍ以上であるので、吸入管と液管との熱交換長さを短くし、冷蔵サイクル用液管あるいは冷凍サイクル用液管に滞留する冷媒の液量を少量に抑制して膨張機構の流量制御を安定して行うことができる。

【０１３５】また、請求項７に記載の発明は、第一の膨張機構と第二の膨張機構は庫内空気と隔離された部分に設置した膨張弁であるので、冷媒漏洩時に冷媒が室内へ漏洩するのを抑制できる。

【０１３６】また、請求項８に記載の発明は、第一の膨張機構あるいは第二の膨張機構を第一の吸入管あるいは第二の吸入管と熱交換する複数のキャピラリで形成し、冷蔵サイクル用液管あるいは冷凍サイクル用液管を複数の前記キャピラリで代用し、複数のキャピラリの流路を切り替えることで抵抗を変化させるので、冷蔵庫の標準的な冷却条件における安定運転時において冷蔵領域冷却時の高蒸発温度化と冷凍領域冷却用膨張機構の入口冷媒乾き度の低下により省エネルギー化を維持しながら、電源投入時に効率良く急冷ができることに加えて、液管を小ボリュームであるキャピラリで代用することで冷媒封入量が低減できる。

【０１３７】また、請求項９に記載の発明は、冷蔵領域と冷凍領域を備えた冷蔵庫であって、圧縮機と、凝縮器と、流路切替弁と、第一のキャピラリと、第二のキャピラリと、第三の蒸発器と、前記第一のキャピラリ及び第二のキャピラリと熱交換する第三の吸入管と、前記第三の蒸発器と前記冷蔵領域内の空気を熱交換する第一の風路と、前記第三の蒸発器と冷凍領域内の空気を熱交換する第二の風路とを備え、前記圧縮機と前記凝縮器と前記流路切替弁と前記第一のキャピラリと前記第三の蒸発器と前記第三の吸入管とを閉ループで構成すると共に、前記第一のキャピラリと並列になるように前記第二のキャピラリとを接続し、前記流路切替弁によりキャピラリへの冷媒の流れを切り替えることにより、前記第一の風路と前記第二の風路を開く時は前記第一のキャピラリを使用し、第二の風路のみを開く時は前記第二のキャピラリを使用して冷媒の流量を可変することを特徴とする冷蔵庫であるので、電源投入時等の過負荷時において抵抗の小さい第一のキャピラリで冷蔵領域と冷凍領域を同時に冷却する冷却運転と冷凍領域のみを冷却する冷却運転を交互に行うことで効率良く急冷が行えると共に、冷蔵領域と冷凍領域を同時に冷却することで冷蔵庫の標準的な冷却条件における安定運転時において冷蔵領域冷却時間を長くして冷蔵領域内の温度変動を抑制できる。

【０１３８】また、請求項１０に記載の発明は、圧縮機は回転数可変型であり、第一の膨張機構と第二の膨張機構は絞り量が変化可能であり、外気温度を検知する外気温センサを有し、前記第一の膨張機構と第二の膨張機構の絞り量は前記外気温センサが検知した外気温から算出した負荷量に相当する必要冷媒流量が流通するように制御され、前記圧縮機の回転数は前記必要冷媒流量から所定蒸発温度になるように制御することを特徴とする請求項４から９のいずれか一項記載の冷蔵庫であるので、電源投入時等の過負荷時において効率良く急冷が行えると共に、常に冷媒流量に対応した熱交換能力を得ることができる蒸発温度となり冷凍サイクルの最大能力を使用して効率良く冷却を行う。

【０１３９】また、請求項１１に記載の発明は、凝縮器と流路切替弁の間に受液器を設けた請求項４から１０のいずれか一項記載の冷蔵庫であるので、低冷媒循環量の冷凍領域冷却から高冷媒循環量の冷蔵領域冷却に切り替わる時の一時的な冷媒循環量不足を解消して早期に冷蔵室冷却の高効率サイクルに移行できる。

【図面の簡単な説明】

【図１】本発明の実施の形態１における冷却サイクル及び冷蔵庫の概略図

【図２】本発明の実施の形態２における冷却サイクル及び冷蔵庫の概略図

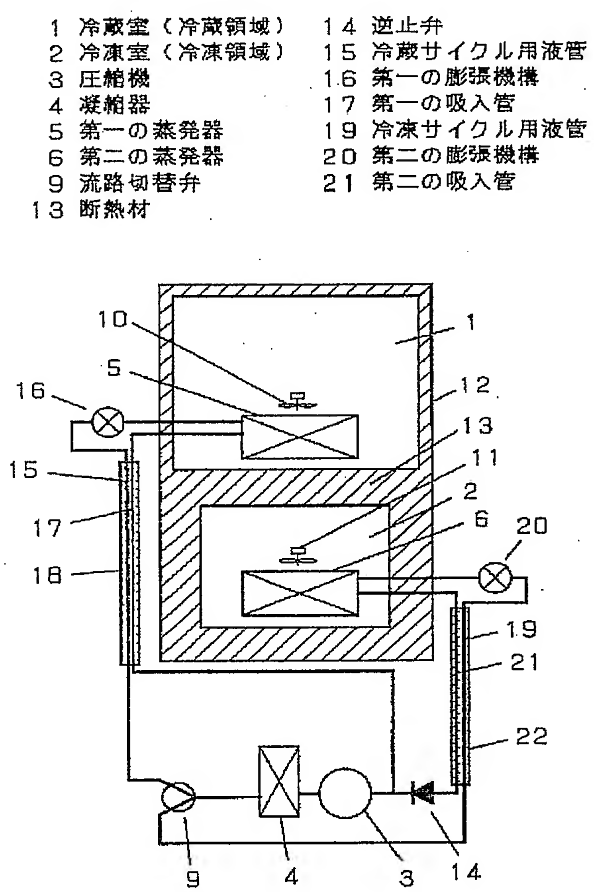
【図３】本発明の実施の形態３における冷却サイクル及び冷蔵庫の概略図

【図４】本発明の実施の形態４における冷却サイクルのＰ－ｈ線図

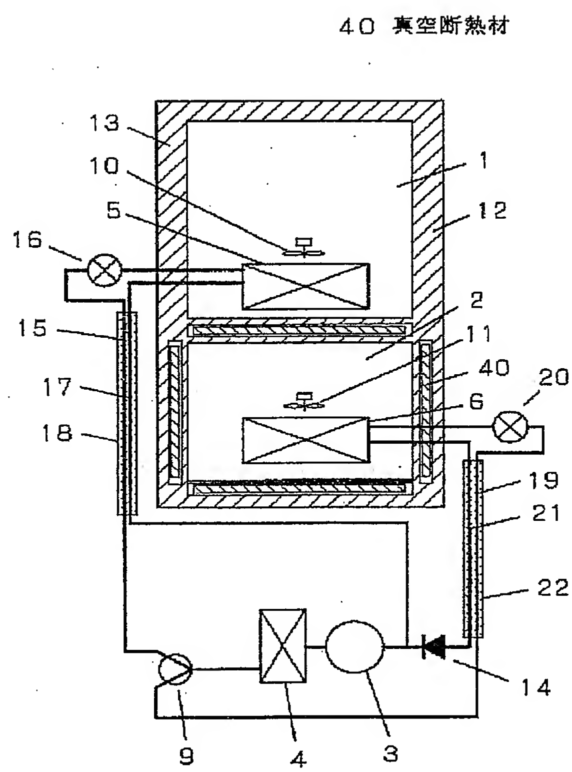
【図５】本発明の実施の形態５における冷却サイクル及び冷蔵庫の概略図

【図６】本発明の実施の形態５における要部の斜視断面図
【図７】本発明の実施の形態６における冷却サイクル及び冷蔵庫の概略図
【図８】本発明の実施の形態７における冷却サイクル及び冷蔵庫の概略図
【図９】本発明の実施の形態８における冷却サイクル及び風路構成の概略図
【図１０】本発明の実施の形態９における蒸発器の蒸発温度と蒸発能力の特性図
【図１１】本発明の実施の形態１０における冷却サイクル及び冷蔵庫の概略図
【図１２】本発明の実施の形態１０における受液器の断面図及び冷蔵庫システムの概略図
【図１３】従来の冷蔵庫の冷却サイクル及び冷蔵庫の概略図
【図１４】従来の冷蔵庫の膨張機構の冷媒流量特性図
【符号の説明】
１　冷蔵室
２　冷凍室
＊３　圧縮機
４　凝縮器
５　第一の蒸発器
６　第二の蒸発器
７　第一のキャピラリ
８　第二のキャピラリ
９　流路切替弁
１３　断熱材
１４　逆止弁
１５　冷蔵サイクル用液管
１６　第一の膨張機構
１７　第一の吸入管
１９　冷凍サイクル用液管
２０　第二の膨張機構
２１　第二の吸入管
３１　第三の蒸発器
３２　第三の吸入管
４０　真空断熱材
４１　受液器

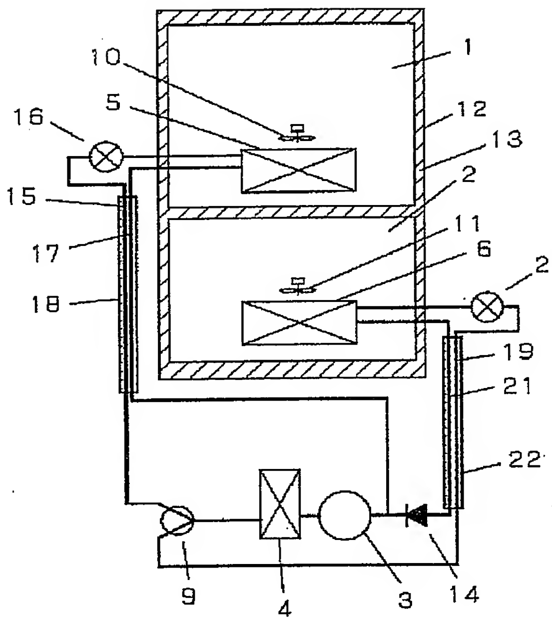
【図１】



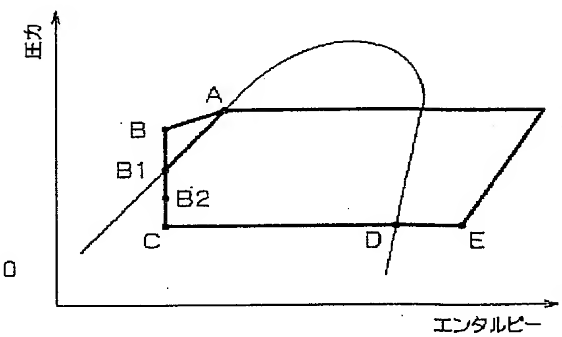
【図２】



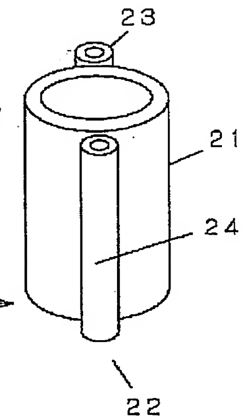
【図３】



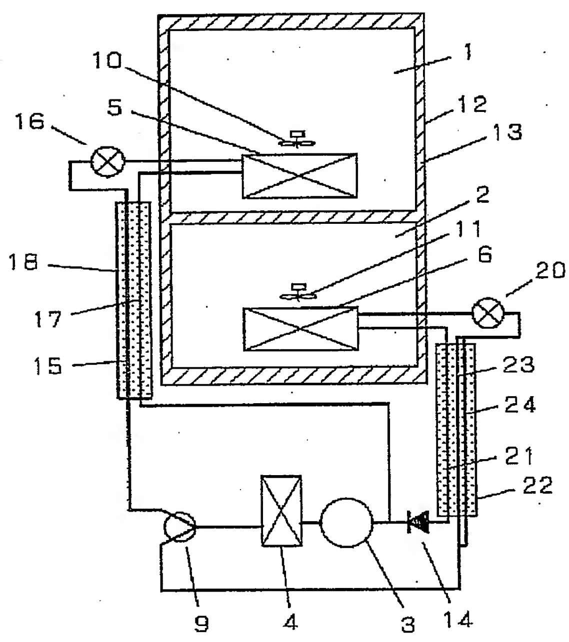
【図４】



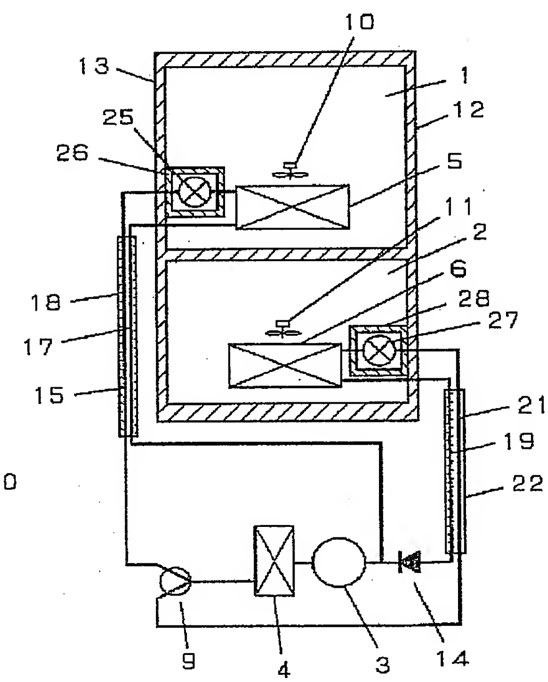
【図６】



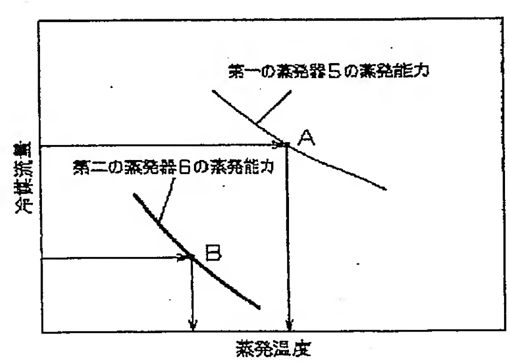
【図５】



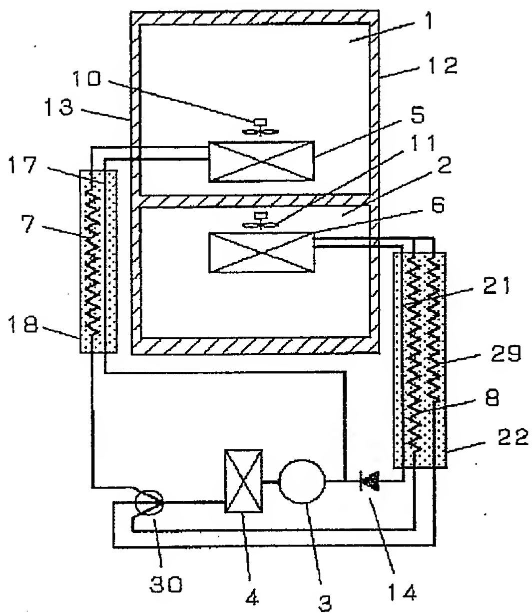
【図７】



【図１０】



【図８】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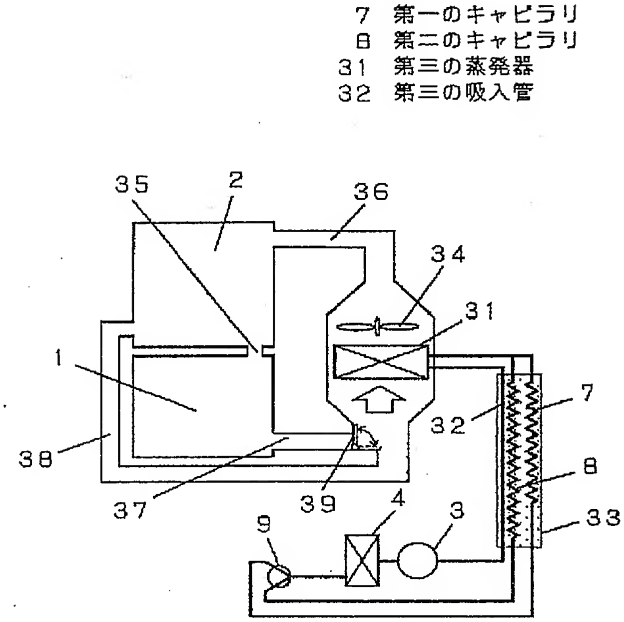
【図９】


7 第一のキャピラリ
8 第二のキャピラリ
31 第三の蒸発器
32 第三の吸入管
35
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34
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1
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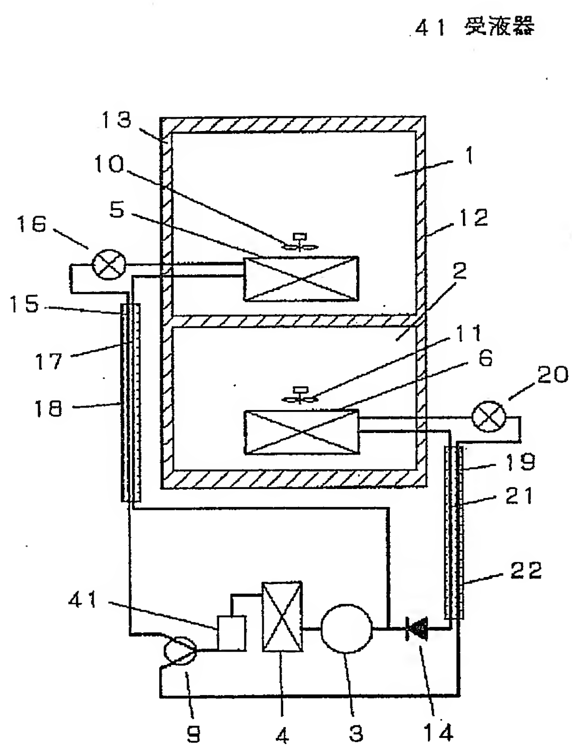
【図１１】


41 受液器
13
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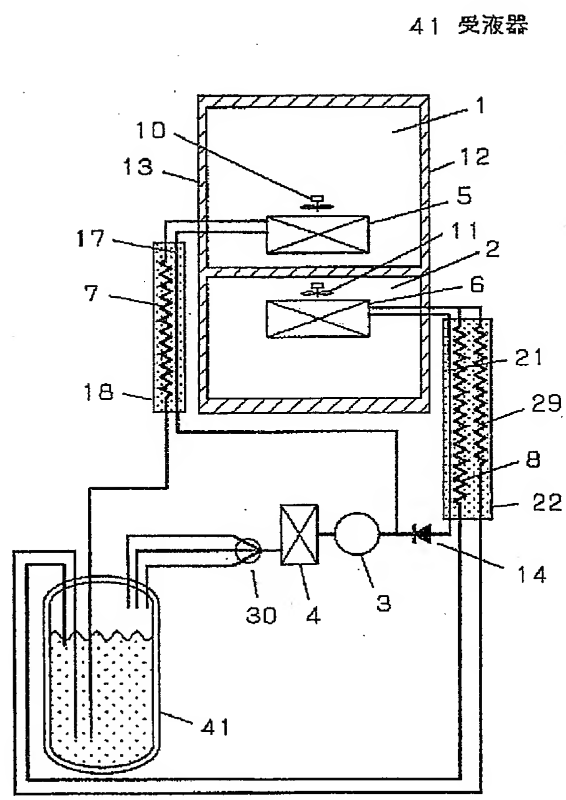
【図１２】


41 受液器
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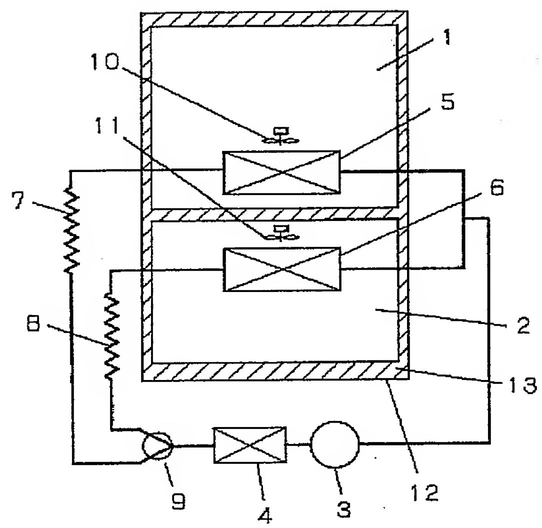
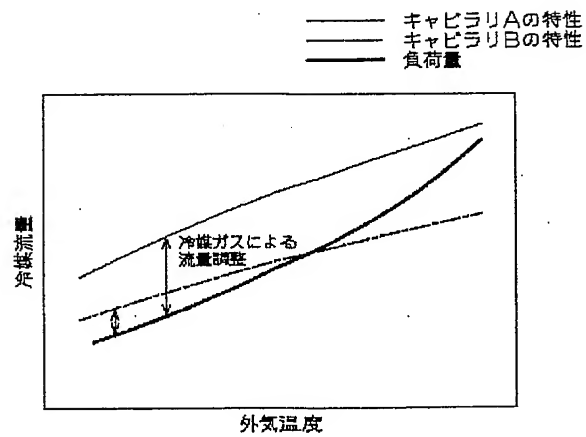
【図１３】



【図１４】



フロントページの続き

Ｆターム(参考)　3L045 AA02 CA03 DA02 EA01 HA02
HA07 JA11 JA15 LA07 NA16
3L102 MB29